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Vice-President: C. L. Kribs, Jr.
Secretary: W. H. Badgett
Treasurer: R. K. Werner
Board of Governors: H. J. Martyn, A. J. Rummel, C. A. McKinney

Washington, D. C.

Headquarters, Washington, D. C.
Meets: Second Wednesday in Month

President: S. P. Eagleton
Vice-President: S. L. Gregg
Secretary: E. V. Fineran
Treasurer: W. R. Lockhart
Board of Governors: Leon Ourusoff, Glegge Thomas, T. H. Urdahl

Wisconsin

Headquarters, Milwaukee, Wis.
Meets: Third Monday in Month

President: J. H. Volk
Vice-President: D. W. Nelson
Secretary: H. C. Frentzel
Treasurer: A. S. Krenz
Board of Governors: M. E. Erickson, C. H. Randolph, Ernest Szekely

TRANSACTIONS
of
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1072

FORTY-FOURTH ANNUAL MEETING, 1938

ESTABLISHING another record of attendance of over 1,000 individuals from all parts of the country and abroad, the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, held at the Hotel Biltmore, New York, N. Y., January 24 to 28, was declared an outstanding success. All of the technical sessions were well attended, with especially large numbers present at the joint sessions held with the *National Warm Air Heating and Air Conditioning Association* and the *American Society of Refrigerating Engineers.*

The Council held two meetings during the week and assisted in the opening of the *Fifth International Heating and Ventilating Exposition* in Grand Central Palace on January 24, which attracted nearly 45,000 people. Representatives from all of the local chapters were present at a conference on January 24 where Society officers and delegates discussed the future activities of the organization as related to local interests.

Numerous technical committee meetings were scheduled where research problems of common interest to the profession were outlined for future consideration. Codes relating to Minimum Requirements for Comfort Air Conditioning and Testing Stoker-Fired Heating Boilers were reported to the Society.

Pres. D. S. Boyden, Boston, Mass., called the 44th Annual Meeting of the Society to order in the ballroom of the Hotel Biltmore and introduced W. E. Heibel, New York, president of the New York Chapter, who welcomed the members and outlined briefly the program of entertainment and important events of the meeting. On behalf of the Society President Boyden responded and thanked Mr. Heibel and the New York Chapter for their cordial greeting.

President Boyden, in reporting the progress during the current year, referred the members to the detailed activities of the Society published in both the JOURNAL and TRANSACTIONS. He then pointed out the rapid growth that has taken place in the industry during the past year and particularly that branch

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referred to as *air conditioning*. Reference to the business conditions throughout the country during the last year were emphasized by the president as being better than for several years, which has contributed to the success of the Society. During the past year the Speakers Bureau of the Society was established, which enabled each Chapter to have various outstanding authorities deliver talks at the local Chapter meetings. According to President Boyden, this service was being used to considerable extent and was being well received. Another outstanding accomplishment of the Society during the past year was the establishment of closer cooperation between the heating, ventilating and air conditioning profession and those associated with the medical and health viewpoints. Much work along this line has already been accomplished by the Society, as pointed out by President Boyden, and there is evidence to indicate that the efforts of this cooperative work will be responsible for the improved health and longevity of mankind.

The Report of the Membership Committee was then presented by R. C. Bolsinger, Conshohocken, Pa.

Report of Membership Committee

Although it is the general thought that a Society like ours should not expand too rapidly, the increased interest in automatic heating, ventilating, and air conditioning demands that we invite those associated, trained, and experienced in these arts to join our organization and to become active members so that the Society as a whole can benefit from their knowledge. Having these ideas in mind, your committee made an extensive drive, the results of which were very gratifying as shown in the accompanying schedule.

Applications Received from January 15, 1937 to January 15, 1938	895
Men Elected in 1937	473
Applications Pending[1]	447

Members Now on Rolls According to Grades

(This group includes 49 Members who were recently Elected)

Members	1466
Associates	692
Juniors	400
Students	35
Life Members	53
Honorary Members	2
Total	2648

Your committee has the following recommendations to make:

1. As the Chapters are the life-blood of the Society, additional Chapters should be formed wherever and whenever the membership is sufficient.
2. The problem of student membership should be considered so as to prove attractive to those who are really interested in heating, ventilating, and air conditioning as we believe this to be for the good of the Society.

The committee takes this opportunity to thank the various Chapter Membership Committees for their very fine cooperation and the Past Presidents, W. H. Driscoll, W. T. Jones, S. R. Lewis, W. H. Carrier, A. C. Willard, H. M. Hart, F. B. Rowley, and J. F. Hale for their contributions regarding membership in the Society.

Respectfully submitted,

MEMBERSHIP COMMITTEE
R. C. BOLSINGER, *Chairman*
ALBERT BUENGER
W. A. RUSSELL

[1] The total number of applications received and elected during 1937 is not the same due to the fact that a few of the men elected in 1937 sent in their applications late in 1936.

A. V. Hutchinson, New York, Secretary, read the Report of the Council.

Report of the Council

The organization meeting of the Council was held in St. Louis January 27, 1937 with Pres. D. S. Boyden presiding, and during the year five meetings were called for the transaction of business. Announcement of the personnel of Standing and Special Committees was made, depositories for funds were selected, a budget was adopted and the appointment of A. V. Hutchinson as Secretary was made.

During the year meetings of the Council were held in St. Louis, Washington, D. C., Swampscott, Philadelphia and New York City.

The Council voted to hold the Semi-Annual Meeting 1938 at Hot Springs, Va., June 20-22, and the 45th Annual Meeting in Pittsburgh, Pa., January 23-25, 1939.

Charters to organize local chapters were approved for the Atlanta Chapter, Golden Gate Chapter at San Francisco, and Iowa-Nebraska Chapter.

A special committee drafted rules of procedure for organizing local chapters and the Council adopted these regulations.

Suggestions for Amendments of the Society's Constitution and By-Laws were received and the Council approved them for submission to members.

The Council voted that dues for 1938 would be $18 for Members and Associates, $10 for Juniors, $3 for Students with initiation fees of $10 for candidates eligible for Member and Associate grades and $5 for those eligible for Junior grade. Approval was given to the transfer of $4100 to the Research Fund in addition to the allotment from membership dues.

Approval was given to the request of the Guide Publication Committee to change the name of THE A.S.H.V.E. GUIDE to THE HEATING, VENTILATING, AIR CONDITIONING GUIDE.

The Council considered the recommendations of a special committee which had studied the needs of local chapters and voted to establish a Speakers Bureau. It was voted to furnish each chapter, upon request, speakers for not more than two meetings between the months of October and April, and also permit the Speakers Bureau to furnish additional speakers when chapters would defray the expense involved.

It was voted that one delegate from each Chapter be brought to the 1938 Annual Meeting and the round trip railroad expense be furnished by the Society.

During the year 16 life memberships were conferred upon the following men: F. C. Black, B. C. Davis, J. A. Donnelly, E. J. Febrey, R. H. Feltwell, R. L. Gifford, J. F. Hale, J. G. Hayes, J. D. Hoffman, H. G. d'Issertelle, J. H. Kitchen, Walter Leek, H. C. Meyer Jr., F. W. Powers, J. N. Russell and G. J. Tobin.

Approval was given by the Council to the revisions sponsored by 17 engineering societies on the Model Registration Law for Professional Engineers and Land Surveyors.

Formal action was taken in accordance with the Constitution and By-Laws on 473 membership applications of which 15 were reinstatements, 48 resignations, and 176 cancellations.

Respectfully submitted,

THE COUNCIL.

The Report of the Secretary was presented by Mr. Hutchinson.

Report of the Secretary

To give a detailed description of the activities carried on at the headquarters office by the Secretary and the staff of eight during the past 12 months would require reams of paper, so that this report will be confined to a summary of the most important items.

The year 1937 was an unusually successful one for the A.S.H.V.E. because of a

phenomenal increase in membership, the organization of chapter activities in three major cities and the extension of research investigations into wider fields.

The volume of work carried on at the headquarters office exceeded any previous year in the history of the Society, because more copies of THE GUIDE were distributed, monthly JOURNALS were larger, two volumes of TRANSACTIONS were issued, and two codes were completed, in addition to the normal routine of handling membership applications, changes of address, employment and other administrative details.

The employment service work has been carried on throughout the year with a notable increase in volume during the last few weeks. Through the chapters, efforts have been made to list those needing employment as rapidly as their needs were made known.

Up to the middle of December, it was evident that 1937 would show a greater increase in membership than had ever been recorded previously. During the last two weeks of the year, the number of applications received almost reached 50 per cent of the total filed during the first 11 months. The total number of applications for the year is 895. On January 1, 1937 there were 2372 members on the rolls and during the year 473 new members were elected; losses were 197, so that the year ended with 2648 members or a net gain of 272.

A heavy addition to the office work was imposed by the establishment of the Speakers Bureau to assist the chapters and from September to December arrangements were completed for 17 speakers to address 19 Chapters.

For those who are statistically minded, it might be noted that over 100,000 letters were mailed and if the stamps purchased were placed side by side they would reach from the New York to New Jersey portals of the Lincoln Tunnel. The number of GUIDES shipped amounted to 30 tons or a car load and one half. The proof reading of the text section of THE GUIDE involved 350,000 words equivalent to four full length novels. The number of GUIDE orders handled amounted to 8000 items. Address changes numbered 1450 and checks issued totaled 725.

During the year assistance was given in the organization of three new chapters with headquarters at Atlanta, Ga., San Francisco, Calif. and Omaha, Nebr.

Your Secretary assisted in the organization of two of these Chapters and visited seven other chapters during the year and also attended the meetings of numerous national organizations interested in A.S.H.V.E. activities.

The reports of the many committees to be presented at this meeting will give an accurate picture of the varied activities of the Society.

It is a great pleasure to acknowledge the cooperation given by the Officers and Council, Chapter Officers and the Committee members, who have assisted in making 1937 a banner year for the Society.

It is fitting that special mention should be made of the splendid service rendered by Mr. James and the other members of the headquarters staff. Their ability to respond to the increased tempo of activities and to handle an extraordinary volume of work deserves special recognition by the members.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Report of the Finance Committee

J. F. McIntire, Detroit, chairman of the Finance Committee, gave the Report of the Society's Certified Public Accountant, showing the conditions of the finances for the calendar year 1937. In commenting on the report, Mr. McIntire referred to the added income of the Society, which came largely from an increase in GUIDE advertising and sales. It was indicated that over 51 firms contributed to the Research Fund during the current year. The chairman of the Finance Committee emphasized the necessity of immediate action when members become delinquent in dues.

Report of Certified Public Accountants

January 19, 1938

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
51 MADISON AVE.
NEW YORK, N. Y.

Gentlemen:

Pursuant to your request we made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y. and the Research Fund for the calendar year ended December 31, 1937 and submit herewith our report.

The work covered a verification of the Assets and Liabilities as of the date previously stated and a review of the operating accounts for the year ended December 31, 1937. For this period recorded cash receipts were traced into the depositories; the cancelled bank vouchers were inspected and compared with the cash records and the Dues Income was accounted for.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1937 and your attention is directed to the following comments thereon:

Cash

Cash on Deposit was verified by direct communication with the banks and reconcilement of the amounts reported to us with the balance shown by the books of the Society.

The Petty Cash kept in the New York Office was verified by count.

A summary of the cash accounts follows:

				—RESEARCH—	
ACTIVE	GENERAL FUND	ENDOWMENT FUND	F. PAUL ANDERSON MEDAL FUND	ENDOWMENT FUND	UNRESTRICTED FUNDS
Bankers Trust Co	$ 9,874.71	$5,322.52	....	....	$ 8,364.23
Chase National Bank, Secretary	3,689.12	....	....	....	....
Chase National Bank, Treasurer	11,250.00	....	....	....	....
Forbes National Bank, Pittsburgh, Pa.	....	....	....	....	806.82
Bowery Savings Bank	8,295.11	....	....	....	....
Central Savings Bank	7,407.48	....	....	....	....
Dry Dock Savings Institution	5,720.86	....	....	....	....
Emigrant Industrial Savings Bank	7,939.50	....	....	....	....
Bank for Savings	....	....	$1,164.61	$491.90	....
Provident Institution for Savings, Boston, Mass.	3,569.91	....	....	....	....
	$57,746.69	$5,322.52	$1,164.61	$491.90	$ 9,171.05
On Hand	100.00	....	....	....	50.00
On Hand for Deposit	42.95	....	....	....	2,385.33
	$57,889.64	$5,322.52	$1,164.61	$491.90	$11,606.38
FROZEN	$ 457.05	....	....	$183.03	....

Marketable Securities

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safe-keeping. No adjustment has been made of the $3,759.96 shrinkage in the market value of these securities, same being included in the attached Balance Sheet at cost.

Accounts Receivable

A list of the Membership Dues Receivable as of December 31, 1937 furnished to us by the management was checked to the individual ledger cards and found in

agreement with the General Ledger Control. The unpaid dues may be summarized as follows:

Dues billed in 1937	$ 6,424.96
Dues billed in 1936	1,910.54
Dues billed in prior years	5,364.56
TOTAL	$13,700.06

Amounts due from GUIDE Advertisers and Other Accounts Receivable verified by trial balance of the individual ledger accounts were found in agreement with the General Ledger Control.

The Reserve for Dues Doubtful of Collection found on the books has been increased by $3,214.25 while the Reserve for Advertisements and Sundry Receivables Doubtful of Collection has been decreased by $443.00. The adjusted Reserves now shown on the accompanying Balance Sheet in our opinion are ample to cover losses resulting from future realization.

Inventories

Items appearing under this caption are based upon quantities submitted by the management and computations made by us. The TRANSACTIONS Inventory priced at cost may be summarized as follows:

Prior Years	$1,681.06
1931	394.62
1932	418.14
1933–34	709.05
1935–36	52.80
TOTAL	$3,255.67

Furniture, Fixtures and Library

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us. We did, however, provide for depreciation of Furniture and Fixtures at the rate of ten (10%) per cent per annum.

Accounts Payable

On December 31, 1937 there remained unpaid invoices amounting to $1,423.57, also the sum of $14,272.70 estimated by the management necessary to complete and make the first mailing of 1938 GUIDE.

Payable to Research Fund

The amounts due the Research Fund representing 40% of Members' and Associates' dues were determined from computations made by us in accordance with Section 5, Article 3, of the By-Laws.

Accrued Accounts

Additional compensation to the Secretary and the clerical staff of the Society has been computed in accordance with the instructions of the Finance Committee.

Reserve for Publications

There has been reserved the sum of $3,500.00 for publication of Volume 43 of TRANSACTIONS. This provision has been made in accordance with Council Authorization.

Deferred Income

Members' Prepaid Dues were ascertained by trial balance of the individual ledger cards. Prepaid Dues from Proposed Members were verified from inspection of the applications found on file.

Reserve for Endowment Fund Adjustment

The original appropriation of the Endowment Fund of $19,641.18 shows an actual reduction in principal of $3,277.90 due to losses incurred from the Sale of Securities. In addition there exists a shrinkage of $3,778.65 in the market value of securities in portfolio. In order to cover these losses and so as to replenish the Endowment Fund the Council on January 27, 1936 authorized the transfer out of the 1935 surplus the sum of $7,500.00. The physical setting aside of either cash or securities has not as yet been consummated.

General Fund

There is attached hereto a schedule showing the changes that have occurred in the General Fund (see p. 10) during the calendar year 1937.

Specific Funds

The specific funds are shown on the Balance Sheet which is shown below after adjustment for interest earned and appropriations authorized by Council.

Operations of the Research Fund for the calendar year 1937 are detailed in schedule attached hereto (see p. 11).

Operations

There is attached hereto Statements of Income and Expenses for the calendar year 1937 showing an excess of Expenses over Income from Society activities of $4,017.72 and a net income of $20,017.73 from GUIDE Operations. In the preparation of these statements 30% of the Salaries and Office Expenses have been allocated to GUIDE Operations.

Respectfully submitted,
TUSA & LA BELLA
FRANK G. TUSA (signed)
Certified Public Accountants

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

December 31, 1937

ASSETS

SOCIETY			
CASH			
On Deposit	$57,746.69		
On Hand	100.00		
On Hand for Deposit	42.95	$57,889.64	
In Closed Bank		457.05	$58,346.69
INVESTMENTS (AT COST)			
Securities (Market Value $11,544.00)		$11,525.31	
ADD: Accrued Interest		61.46	11,586.77
ACCOUNTS RECEIVABLE			
Membership Dues	$13,700.06		
LESS: Reserve for Doubtful	8,315.27	$ 5,384.79	
Advertisers and Sundry Debtors	$33,631.54		
LESS: Reserve for Doubtful	1,521.12	32,110.42	37,495.21
INVENTORIES			
Transactions		$ 3,255.67	
Emblems and Certificate Frames		34.16	3,289.83
PERMANENT			
Library		$ 300.00	
Furniture and Fixtures	$ 6,056.66		
LESS: Reserve for Depreciation	5,080.92	975.74	1,275.74
DEFERRED CHARGES			
Meeting Expenses			216.73 $112,210.97

SPECIFIC FUNDS			
ENDOWMENT FUND			
Cash on Deposit		$ 5,322.52	
Securities at Cost (Market Value $8,140.00)	$11,918.65		
ADD: Accrued Interest	111.35	12,030.00	17,352.52
F. PAUL ANDERSON AWARD FUND			
Cash on Deposit			1,164.61
RESEARCH FUND			
ENDOWMENT FUND			
Cash on Deposit	$ 491.90		
Cash in Closed Bank	183.03	$ 674.93	
UNRESTRICTED FUND			
Cash on Deposit	$ 9,171.05		
Cash on Hand	50.00		
Cash on Hand for Deposit	2,385.33	11,606.38	12,281.31
			$143,009.41

LIABILITIES AND CAPITAL

SOCIETY			
ACCOUNTS PAYABLE		$15,696.27	
PAYABLE TO RESEARCH FUND			
On Dues as and when Collected		3,318.74	
ACCRUED ACCOUNTS			
Compensation—Secretary and Staff		2,579.79	
RESERVE FOR PUBLICATION			
TRANSACTIONS, Vol. 43		3,500.00	
DEFERRED INCOME			
Prepaid Dues—Members	$ 1,396.16		
Prepaid Dues—Proposed Members	2,134.55	3,530.71	
RESERVE FOR ENDOWMENT FUND REPLENISHMENT		7,500.00	
TOTAL LIABILITIES		$36,125.51	
GENERAL FUND			
Society		76,085.46	$112,210.97
SPECIFIC FUNDS			
ENDOWMENT FUND			
Principal		$16,363.28	
Unexpended Income		989.24	17,352.52
F. PAUL ANDERSON AWARD FUND			
Principal		$ 1,000.00	
Unexpended Income		164.61	1,164.61
RESEARCH FUND			
ENDOWMENT FUND			
Principal	$ 600.00		
Unexpended Income	74.93	$ 674.93	
UNRESTRICTED FUND			
Balance—January 1, 1937	$15,060.16		
Excess of Expenses over Income for the Year Ended December 31, 1937	3,453.78	11,606.38	12,281.31
			$143,009.41

NOTE "A"—This Balance Sheet is subject to comments contained in the letter attached to and forming a part of this report.

STATEMENT OF INCOME AND EXPENSES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Calendar Year Ended December 31, 1937

INCOME

FROM MEMBERS			
DUES—RENEWALS			
Members and Associates		$33,606.00	
LESS: Cancellations	$1,881.50		
Provision for Uncollectable	3,214.25	5,095.75	
		$28,510.25	
LESS: 40% to Research Laboratory		12,689.80	
		15,820.45	
Junior and Student Members		3,410.00	$19,230.45
DUES—NEW MEMBERS			
Members and Associates	$4,939.00		
LESS: 40% to Research Laboratory	1,975.60	$2,963.40	
Junior and Student Members		973.55	3,936.95
			$23,167.40
INCOME FROM SALES OF EMBLEMS AND CERTIFICATE FRAMES			53.07
TOTAL INCOME FROM MEMBERS			$23,220.47
FROM OTHER SOURCES			
Editorial Contract		$14,131.43	
LESS: Members' Subscriptions		5,058.79	$9,072.64
Net Income—Reprints, Books and Codes			169.95
Net Income—TRANSACTIONS—Per Schedule			704.32
TOTAL INCOME FROM OTHER SOURCES			9,946.91
FROM INVESTMENTS			
Interest from Current Funds			$668.07
Interest from Securities			305.00
TOTAL INCOME FROM ALL SOURCES			$34,140.45

EXPENSES

APPORTIONABLE EXPENSES			
Salaries—Secretary and Staff		$17,670.15	
Traveling—Secretary and Staff		1,291.45	
Rent and Light		2,589.96	
Telephone		641.37	
Telegrams		306.38	
Postage		2,393.45	
Depreciation—Furniture and Fixtures		201.55	
General Printing		911.93	
Office Supplies		269.02	
Multigraphing and Stencils		635.38	
Professional Services		400.00	
Bank Charges		55.72	
Miscellaneous		736.72	
		$28,103.08	
LESS: 30% Applicable to GUIDE Operations		8,430.92	$19,672.16
			$19,672.16 $34,140.45
OTHER EXPENSES			
Meetings—Annual and Semi-Annual		$ 1,857.37	
President's Fund		1,145.63	
Yearbook		1,250.11	
Awards, Membership Certificates, Medals, etc		288.10	4,541.21
OTHER DEDUCTIONS			
Chapter Meetings Allowance		$1,000.00	
Provision for 1937 TRANSACTIONS		3,500.00	
A.S.H.V.E.—*A.S.R.E.* Joint Committee		898.27	
Contribution to *A.S.A.*		100.00	
Council and Chapter Delegates Travel		3,497.25	
Speakers Bureau		849.28	
Appropriations to Research Fund		4,100.00	13,944.80 38,158.17
EXCESS OF EXPENSES OVER INCOME FROM SOCIETY ACTIVITIES			$ 4,017.72

ANALYSIS OF GENERAL FUND

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Calendar Year Ended December 31, 1937

GENERAL FUND—JANUARY 1, 1937—PER FORMER REPORT			$60,055.18
ADDITIONS			
Income from GUIDE Operations	$20,017.73		
LESS: Excess of Expenses over Income from Society Activities	4,017.72	$16,000.01	
Additional Advertising Income from Publication of 1937 GUIDE		50.50	
Partial Collection of GUIDE Advertising Accounts Previously Written Off		2.77	16,053.28
			$76,108.46
DEDUCTIONS			
Adjustment of Dues Accounts of the following per Council Minutes of April 9, 1937			
H. K. Hicks		$ 5.00	
R. E. Poehner		18.00	23.00
GENERAL FUND—DECEMBER 31, 1937—PER BALANCE SHEET			$76,085.46

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Calendar Year Ended December 31, 1937

	Actual	*Budget Provision*	*Increases Decreases*
INCOME			
Membership Dues	$ 23,167.40	$23,000.00	$ 167.40
Editorial Contract	14,131.43	13,500.00	631.43
Profit—Sales of Emblems and Certificates	53.07	100.00	46.93
Profit—Reprints and Books	309.71	300.00	9.71
Sales TRANSACTIONS	704.32	800.00	95.68
Interest—Savings Banks and Securities	973.07	1,000.00	26.93
GUIDE—Advertising	34,418.78	31,000.00	3,418.78
GUIDE—Copy Sales	31,877.37	30,000.00	1,877.37
	$105,635.15	$99,700.00	$ 5,935.15
EXPENSES			
General Society Activities			
Meetings	$ 1,857.37	$ 1,600.00	$ 257.37
Chapter Allowance for Meetings	1,000.00	1,000.00	....
Promotional	....	3,000.00	3,000.00
Speakers Bureau	849.28	1,000.00	150.72
Subscriptions (H.P. & A.C.)	5,058.79	5,000.00	58.79
Postage	2,393.45	2,500.00	106.55
Rent	2,589.96	3,000.00	410.04
TRANSACTIONS	3,500.00	3,500.00	....
Codes (Net of Sales)	139.76	500.00	360.24
Salaries (Includes Additional Compensation)	17,670.15	19,000.00	1,329.85
General Printing	911.93	800.00	111.93
Membership Certificates and Emblems	288.10	600.00	311.90
President's Fund	1,145.63	1,800.00	654.37
Travel—Secretary	1,291.45	1,500.00	208.55
Council and Chapter Delegates Travel	3,497.25	4,400.00	902.75
Bank Charges and Discounts	55.72	100.00	44.28
Multigraphing	635.38	500.00	135.38
Telephone	641.37	500.00	141.37
Telegraph	306.38	300.00	6.38
Professional Services	400.00	500.00	100.00
General Office	736.72	600.00	136.72
Office Supplies	269.02	500.00	230.98
Furniture and Fixtures	201.55	500.00	298.45
Special Appropriations	998.27	550.00	448.27
	$ 46,437.53	$53,250.00	$ 6,812.47
COST OF GUIDE			
Editorial and Advertising Salaries	$ 6,286.64	$ 7,200.00	$ 913.36
Committee Travel	251.01	750.00	498.99
Printing and Binding	15,829.26	18,500.00	2,670.74
Mailing	1,575.00	2,000.00	425.00
Paper	4,546.96	4,000.00	546.96
Cuts	514.32	500.00	14.32
Advertising Sales Promotion	1,452.61	2,000.00	547.39
Copy Sales	7,391.70	7,500.00	108.30
Yearbook Section	1,250.11	1,000.00	250.11
	39,097.61	43,450.00	4,352.39
	$ 85,535.14	$96,700.00	$11,164.86
Unbudgeted—Appropriations to Research Fund	4,100.00	....	4,100.00
	$ 89,635.14	$96,700.00	$ 7,064.86

BUDGET COMPARISON—RESEARCH FUND

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Calendar Year Ended December 31, 1937

	Actual	*Budget Provision*	*Increases Decreases*
INCOME			
Balance from 1936 Committee—Cash in Pittsburgh and New York Banks	$15,060.16	$12,088.23	$2,971.93
Dues—Current and Prior Years	13,744.19	13,800.00	55.81
Interest on Bank Balances	....	30.00	30.00
Interest on Savings Account	59.76	70.00	10.24
Interest on Society Endowment	....	600.00	600.00
Contributions—General	5,200.00	10,000.00	4,800.00
Contributions—Earmarked	500.00	500.00	....
Special Council Appropriation	2,500.00	4,000.00	1,500.00
Special Council Appropriation	200.00	200.00	....
Special Council Appropriation	1,400.00	1,400.00	....
Contributions—"Glass" Study	3,650.00	4,075.00	425.00
	$42,314.11	$46,763.23	$4,449.12
EXPENSES			
Travel—Committee and Laboratory Staff	$ 1,707.82	$ 2,000.00	$ 292.18
Salaries—Laboratory Staff	13,296.00	15,000.00	1,704.00
Salaries—Student Help	1,700.55	1,800.00	99.45
Cooperative Research—Earmarked	731.25	731.25	....
Cooperative Research—General	2,074.22	5,400.00	3,325.78
Cooperative Research—Medical School	2,025.00	2,100.00	75.00
Laboratory Supplies and Equipment	692.56	700.00	7.44
Office Supplies and Equipment	518.13	500.00	18.13
Printing	....	400.00	400.00
Exhibits	....	200.00	200.00
Correlating Thermal Research	1,260.00	1,500.00	240.00
Salary—Office of Chairman	275.75	900.00	624.25
Promotion and Publicity	5,091.16	4,400.00	691.16
Meetings	161.19	300.00	138.81
Contingency Fund	....	500.00	500.00
N. D. H. A. Exhibit	211.03	200.00	11.03
Special Study on "Glass"	963.07	2,125.00	1,161.93
	$30,707.73	$38,756.25	$8,048.52
UNAPPROPRIATED BALANCE	11,606.38	8,006.98	3,599.40
	$42,314.11	$46,763.23	$4,449.12

NOTE.—The above Budget Comparison includes all revisions voted by the Committee on Research during the calendar year 1937. We note that in preparing the 1937 budget the Research Committee has shown the cash balance taken over from the 1936 Research Committee at $12,088.23 due to the exclusion of the cash on deposit with the Bank for Savings.

The Report of the Tellers was then presented by John James, New York, as follows:

Reports of Boards of Tellers

BALLOT FOR OFFICERS

	For
President—E. Holt Gurney	700
First Vice-President—J. F. McIntire	702
Second Vice-President—F. E. Giesecke	702
Treasurer—A. J. Offner	702
Members of the Council—Three-Year Term	
N. D. Adams	702
A. P. Kratz	702
J. H. Walker	702
G. L. Wiggs	701
Total Ballots Counted	734
Ballots Invalid	32

Scattering votes were recorded for F. J. Linsenmeyer, F. C. Hibbs, and A. W. Luck.

BALLOT FOR COMMITTEE ON RESEARCH

Three-Year Term

M. K. Fahnestock	702
W. L. Fleisher	700
R. J. Tenkonohy	702
T. H. Urdahl	701
B. M. Woods	702
Total Ballots Counted	734
Ballots Invalid	32

Scattering votes were recorded for Tom Brown and C. A. Thinn.

BOARD OF TELLERS
R. V. SAWHILL, *Chairman*
J. R. MURPHY
R. G. OLSON

BALLOT FOR AMENDMENT TO CONSTITUTION AND BY-LAWS

ARTICLE C-II, SECTION 2

Article C-II, *Section* 2—The membership of the Society shall consist of Honorary Members, Life Members, Members, Associate Members, Junior Members, and Student Members.

Number of Ballots *For*	666
Number of Ballots *Against*	22
Total Number of Ballots	688
Illegal Ballots	33

BOARD OF TELLERS
M. C. BARNUM, *Chairman*
E. C. BLACKBURN, JR.
G. A. DORNHEIM

BALLOT FOR CODE OF MINIMUM REQUIREMENTS FOR COMFORT AIR CONDITIONING

Number of Ballots *For*	484
Number of Ballots *Against*	31
Total Number of Ballots	570
Illegal Ballots	55

BOARD OF TELLERS
C. L. CHASE, *Chairman*
C. S. PABST
C. D. SHIELDS

The second session was a joint meeting with the *National Warm Air Heating and Air Conditioning Association,* held in the ballroom of the Hotel Biltmore, with Pres. D. S. Boyden and Pres. W. L. Rybolt, Ashland, Ohio, of *N.W.A.H.A.C.A.* presiding jointly as chairmen.

President Boyden opened the third session and first called for the Report of the Treasurer, presented by A. J. Offner, New York.

Report of the Treasurer

When one sees the many activities being carried on at these meetings, when one receives THE GUIDE and other publications, when one notes the excellent and important work being carried on by our Research Committee, one realizes that it not only takes much work but also much money. We would therefore like to appeal to the members to pay their dues promptly, not only so that the necessary money is available, but also in order that we do not have the unnecessary expense of billing such members.

We also have another problem and that is the collection of dues which are past due. While we appreciate and understand the reason in many cases, and while we try to be as lenient and considerate as we possibly can, we are forced by the require-

ments of our Constitution to drop such members. We are very reluctant to do so and we would also like to appeal to the help of the members in such cases.

W. A. Danielson, Fort Knox, Ky., chairman of the Committee on Research, was then introduced and abstracted the Report of the Committee on Research for the year 1937 which he and the Director of the Research Laboratory, F. C. Houghten prepared.

Report of the Committee on Research

Entering its nineteenth year of service to the heating, ventilating and air conditioning industries, the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS finds itself in a more strategic position than ever before as a coordinator of basic investigations, and as the medium through which vital and unbiased studies of fundamental aspects of these sciences are carried on.

The phenomenal rise of air conditioning to a position of such prominence in the public mind that it is a factor in almost every building operation, enters into every plan for store modernization or industrial plant rehabilitation and is considered by almost every home owner or prospective home owner, has brought with it new opportunities for service, and greater responsibilities.

Today's concepts of air conditioning stem directly from the basic work of the A.S.H.V.E. Laboratory, which for more than a dozen years has constantly been adding to the fundamental knowledge of the art upon which a great industry is being built. In a very real sense therefore, scientific progress in air conditioning and the older arts of heating and ventilating out of which it grew, is synonymous with the growth and development of the research sponsored by this Society.

Out of the work of the past year, there has emerged a clearer realization of the problems which lie ahead and a sharpened conception of the function which the Society can perform in guiding the industry along sound lines. The multiplicity of channels through which air conditioning and related arts is developing and its rapid growth and quickening tempo has substantially increased the complexity of the research problems seeking solution, while at the same time bringing many new subjects forward for investigation.

In these pages are presented brief summaries of the activity of twenty-one technical committees cooperating with the Committee on Research, each investigating some fundamental question of heating, ventilating or air conditioning upon which either new knowledge is required or the improvement of existing data and its revision in the light of new conditions, has become necessary.

That aspect of the problem as a whole which concerns itself with the fundamental human reaction to, and the effect of air conditioning on, human beings, has again been a chief concern of the Committee on Research. In particular, it is meeting a well-defined need for basic information as to the comfort requirements in summer cooling, in order that commercial engineering developments may be directed with more exact knowledge of conditions to be met in practice.

As a natural outgrowth of the continuous study of the physiological reactions of normal people to air conditioning, problems in the application of air conditioning to the treatment of disease are rapidly coming to the fore.

Investigation and study of the therapeutic value of air conditioning has received greater emphasis and it is gratifying to note that the medical profession is becoming more broadly interested in air conditioning than at any time. Evidence of a better understanding and an attitude of mutual respect and cooperation between the physician and the engineer has been especially marked during the year.

A third important aspect of research has been further determination of the effect of air conditioning on the performance of building materials and the extension of this research to the development of new data with respect to various forms of glass insulating and glass building materials which are coming into wider use.

In the division dealing with performance standards, the activity of the year has been concerned with testing the frictional resistance to the flow of air in small ducts, the heat transfer performance of finned-tubes with forced air circulation, direct and indirect methods of radiation with gravity air circulation, the performance of

equipment for residence air conditioning, efficiency of air cleaning devices, and corrosion in heating and air conditioning systems, to mention but a few of the subjects investigated.

Looked at as a continuous and progressive program, the year 1937 was marked by the collection of important data which were the means of laying the groundwork for more extensive research during the coming year. It is of interest to note that upwards of 250 individuals, both within and without the membership of the Society, were engaged during the year in the work of the various technical committees and that a number of technical papers were prepared bearing on work initiated by the Committee or carried out in cooperation with engineering schools, colleges and other technical bodies.

Distinguished members of their professions and experts in numerous lines of technical activity have given unstintingly of their time to maintain A.S.H.V.E. Research on the high plane which has characterized it for nearly two decades.

There is gratifying evidence of a broader interest on the part of the industry in general to the importance of A.S.H.V.E. Research, which may be attributed in no small measure to the educational publicity activities carried on during the year.

Technical Advisory Committee OH-22—COMFORT AIR CONDITIONING—C. Tasker, *Chairman;* A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington, R. E. Keyes, A. B. Newton, C. P. Yaglou.

The work of this committee for the past year included a study of skin temperatures and comfort as they may be affected by a "draft" or air movements of different velocity and temperature, and a study of the relation between a person's feeling of warmth and the effective temperatures maintained in summer air conditioned space. This study shows a definite drop in the skin temperature of the parts affected, varying with the velocity and temperature of the air.

The study of the relation between comfort and the effective temperature maintained in summer air conditioned space was continued in Pittsburgh, Toronto and Texas, somewhat along the lines followed during 1936. A very comprehensive study was made of the reactions of 275 office workers of the Minneapolis-Honeywell Regulator Co., in Minneapolis, to the air conditions maintained throughout the past summer. The small difference between the results found in this study and those previously reported indicates the need for continuing the study in a number of other geographical locations, in order to establish the effects which such variations may have on the conditions under which people may be most comfortable. A paper, Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, resulting from the Minneapolis study, will be presented at the Annual Meeting.

Technical Advisory Committee OH-3—RELATION OF BODY CHANGES TO AIR CHANGES—Dr. E. Vernon Hill, *Chairman;* N. D. Adams, J. J. Aeberly, John Howatt, A. P. Kratz, P. J. Marschall, V. L. Sherman.

A comprehensive program for a study of the application of air conditioning in the treatment of certain diseases was developed about two years ago and a study initiated by Dr. Robert Keeton of the College of Medicine of the University of Illinois, at Chicago. Observations have been made throughout the year, and a paper (Physiologic Response of Man to Environmental Temperature, by Dr. F. K. Hick, Dr. R. W. Keeton and Nathaniel Glickman) is being presented before the Society at the Annual Meeting in New York.

Technical Advisory Committee C-24—CLIMATE AND AIR CONDITIONING—Dr. C. A. Mills, *Chairman;* Major G. C. Dunham, James Govan, Dr. W. J. McConnell, C. F. Neergaard, Dr. F. M. Pottenger, Jr., E. L. Weber, C.-E. A. Winslow.

This committee has been interested in developing a comprehensive program for a study of the effect on persons of various climatic changes in the atmosphere.

It is also continuing the collection of basic data relating to physiological response to physical environment including investigation of the use of radiant heat for comfort air conditioning—both winter and summer. The physiological effects of storm fluctuations in barometric pressure are also being studied.

Few technical advisory committees organized under the Committee on Research, are operating on as broad a base as the Committee on Climate and Air Conditioning.

Technical Advisory Committee C-17—TREATMENT OF AIR WITH ELECTRICITY—C.-E. A. Winslow, *Chairman;* R. D. Bennett, W. H. Carrier, L. W. Chubb, Major W. D. Fleming, R. F. James, L. R. Koller, Dr. C. A. Mills, Dr. E. B. Phelps, G. R. Wait, W. T. Wells.

This is an important committee which is at present reviewing research activity dealing with the application of electricity to the treatment of air for human comfort and health and encouraging work by its members and others.

At present, the most promising approach to this problem appears to be along the lines of changes in the atmosphere due to irradiation which seem to be associated with the stimulation of metabolic activity.

It is the opinion of the committee that the destruction of bacteria in the air by direct application of ultra-violet light and the exclusion of bacteria by a vertical curtain of irradiation have passed the experimental stage and give real promise in hospitals (operating rooms, communicable disease wards, wards for infants) and in certain food industries.

The committee feels that Professor Phelps' studies of the possible effects of irradiation upon bacterial dissociation and mutation are of great interest and hopes it may receive a report from him on this matter.

It is believed by the committee that electrostatic precipitation as a means of removing suspended matter (dust particles and microbes) from air admitted to an occupied space presents great promise.

Technical Advisory Committee IF-31—WEATHER DESIGN CONDITIONS—W. E. Stark, *Chairman;* E. W. Goodwin, A. C. Grant, J. H. Kincer, A. P. Kratz, L. S. Ourusoff.

This committee was recently appointed to outline plans for analysis of Weather Bureau data, with a view of establishing a better understanding of the design outside temperatures for both winter and summer air conditioning. This committee is also interested in a better understanding of the "degree-day" application to heating and cooling problems.

This research was initiated partially in response to a need expressed by the Air Conditioning Code Committee of the A.S.H.V.E., working jointly with the *A.S.R.E.*

This committee represents an organized effort to make an intensive study of natural weather conditions as affecting design and may eventually influence the form or substance of data now given out by the United States Weather Bureau.

Technical Advisory Committee—HOSPITAL AIR CONDITIONING—Dr. T. L. Hazlett, *Chairman;* Dr. C. J. Barone, Dr. B. Z. Cashman, Dr. M. B. Ferderber, C. M. Humphreys, G. S. McEllroy, H. W. Redding, Dr. A. W. Sherrill.

Through cooperation between the Research Laboratory and the Department of Medicine, University of Pittsburgh, a program has been developed for a study of factors entering into the air conditioning of an operating room and a recovery ward. In this connection, an experimental air conditioning system has been provided by the interested industry and installed in connection with an operating room and recovery ward in the Magee Hospital, one of a group of hospitals connected with the University of Pittsburgh Medical School. The installation is near completion, and it is hoped that the University of Pittsburgh, in cooperation with the Laboratory, will be able to collect data throughout the coming year.

The program includes a study of:

(1) Optimum temperature and humidity conditions within the operating room and recovery ward.
(2) The bacteria content of the atmosphere in an operating room as operated in the past, and as operated with the air conditioning system.
(3) The relation between past operation without air conditioning and future operation with air conditioning and controlled ventilation, as it affects the concentration of anesthetic at different points with respect to the operating table.

Technical Advisory Committee IP-4—AIR CLEANING—H. C. Murphy, *Chairman;* M. I. Dorfan, C. E. Lewis, S. R. Lewis, G. W. Penney, A. L. Simison, W. O. Vedder.

The work of this committee throughout the year has consisted in keeping in contact with and, wherever possible, correlating the efforts of different groups interested

in developments in the field of air cleaning. Notable work by Dr. C. B. Coulter and Dr. F. M. Stone, of the College of Physicians and Surgeons, Columbia University, as reported in a recent issue of the *American Journal of Public Health*, indicates that an average of 16 times as many bacteria are found in unventilated public telephone booths in New York City as in well ventilated booths in the air conditioned building at Radio City, where filtered air is supplied.

Plans for work of the committee during the coming year include analyses of air-borne bacteria found in different types of buildings and their relation to air conditioning applications in general, and methods of filtering or cleaning the air in particular.

Technical Advisory Committee C-12—TRANSPORTATION AIR CONDITIONING—L. B. Miller, *Chairman;* Dr. T. R. Crowder, F. B. Rowley, A. E. Stacey, Jr., L. W. Wallace.

This committee, appointed during the year, is organizing with a view to developing a comprehensive study of the application of air conditioning to railroads.

The principal purpose of this committee is to cooperate along a broad front with interests identified with railroad operation in order to assist in establishing correct comfort requirements for railroad car air conditioning.

Technical Advisory Committee OH-5—GENERAL AIR CONDITIONING COMFORT REQUIREMENTS—C. P. Yaglou, *Chairman;* J. J. Aeberly, Dr. R. R. Sayers, C.-E. A. Winslow.

Research in the general field of comfort and health has continued during the year with renewed vigor and with considerable expansion in all directions. The Research Laboratory in Pittsburgh and cooperating institutions continued several important studies on drafts and summer cooling, as reported elsewhere by other committees.

The Harvard School of Public Health, in cooperation with the Research Laboratory, published a paper (Ventilation Requirements, Part II, by C. P. Yaglou and W. N. Witheridge, A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 423), dealing with disappearance of ventilation odors. The evidence presented indicates that the importance of per capita air space on minimum ventilation requirements for the control of body odors arises from the fact that such odors are not stable, but tend to disappear rapidly of their own accord.

Large rooms, it has been found have an advantage over small ones, as they act like reservoirs, allowing body odor to disappear with a minimum of outdoor air supply. The odor of tobacco smoke, on the other hand, not only remains longer in the air, but its intensity increases considerably as the smoke gets stale.

It would appear from such observations as the committee has made to date, that smoking rooms should be made as small as practical and ventilated rapidly. The broad scope of this committee is such that it can to a certain extent, correlate the work of other committees engaged in various phases of comfort air conditioning, although its primary function has been the determination of proper outside air quantities requisite for good indoor air requirements.

Technical Advisory Committee IP-9—RADIATION WITH GRAVITY AIR CIRCULATION —M. K. Fahnestock, *Chairman;* B. C. Benson, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson, R. N. Trane, T. A. Novotney.

Work under this committee was continued during the year at the University of Illinois in cooperation with the Research Laboratory. A paper (The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick, A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 389), was presented to the Society at its Semi-Annual Meeting at Swampscott.

Under the program of the committee to be carried on during 1938-39, the following research should be divided into two parts: (I) That which may be done in the warm wall test booth; and (II) that which may be done in the room heating testing plant.

Under (I) it is hoped that the study of the inlet and outlets of convector heater cabinets with reference to their effect upon the heat output or capacity of the units may be completed in 1938.

Under (II) it is expected that the remodeling of the room heating testing plant will be completed and operations started.

(1) The first tests will be to correlate the performance of conventional tubular steam radiators (sections on 2½ in. centers) in the new 15 ft x 18 ft test room with their performance in the small 9 ft x 11 ft test rooms which were in the old plant.
(2) Tests to be made with conventional 3-tube radiators standing in the room 2½ in. from the plastered surface and placed in recesses below the windows in the exposed walls.
(3) Tests with several different sizes of the new light-weight type of direct steam radiators (sections on 1½ in. centers) to determine their performance, including capacities and the manner in which they distribute heat in the room.

Technical Advisory Committee IP-10—HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION—F. B. Rowley, *Chairman;* H. F. Hutzel, R. H. Norris, C. H. Randolph, W. E. Stark, G. L. Tuve, C. F. Wood.

Work under this committee was carried on at Case School of Applied Science, in cooperation with the Research Laboratory. In a paper (Performance of Fin Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman, A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 367) the authors indicated the importance of developing a Code for the testing and rating of dehumidifying coils.

During the year, L. J. Seigel, on a research fellowship, succeeded in working out a very promising analysis which has two important advantages over any other method which has been proposed to date:

(1) Any of the common types of practical problems can be solved directly by formula, without trial and error.
(2) The entire range of performance of the coil can be obtained by running dry-cooling tests only.

A comparative study was made of four methods of computing the performance of a dehumidifying coil, including those published during 1936 by Goodman and by Wells, indicating the advantages of the new method, and steps were taken to check it by a series of tests on one of the coils not previously tested.

Technical Advisory Committee IP-30—COOLING LOAD IN SUMMER AIR CONDITIONING—J. H. Walker, *Chairman;* C. M. Ashley, John Everetts, Jr., F. H. Faust, H. F. Hutzel, L. S. Morse, A. E. Stacey, Jr., R. M. Strikeleather.

This committee was appointed during the year to study particularly the cooling load in summer cooling and air conditioning for human comfort, resulting from the increased heat transfer through walls and windows due to sun radiation. A comprehensive program was developed with the hope that it could be carried out during the past summer.

This investigation included a study of the heat transfer through 12 sample walls, to be built and studied at the Research Laboratory. These walls were to be subject to sun radiation on the south side and means were to be provided for measuring the heat transfer through the inside surface by a modification of the guarded hot box method.

Technical Advisory Committee IP-21—AIR DISTRIBUTION—Ernest Szekely, *Chairman;* S. H. Downs, M. K. Fahnestock, F. J. Kurth, D. W. Nelson, C. H. Randolph, J. E. Schoen, G. L. Tuve, J. H. Van Alsburg.

This committee, which has been organized and has had plans available for a comprehensive study for several years, found itself for the first time in possession of sufficient funds to make any material progress during the past year. A study was initiated by Prof. G. L. Larson of the University of Wisconsin in cooperation with the Society's Laboratory and the Barber-Colman Co., and an investigation of some factors dealing with horizontal distribution of air into an enclosure was made. The study as outlined deals particularly with the relation between the air distribution in the occupied zone to (1) the distance of discharge below ceiling; (2) the aspect ratio of the opening; and (3) the angularity of the blades in grilles. One paper, Air Distribution from Side Wall Outlets, resulting from the work, has been released for publication and presentation at the Annual Meeting, showing some notable progress.

XUM

Technical Advisory Committee IP-20—INTERMITTENT HEATING—E. K. Campbell, *Chairman;* W. L. Cassell, E. F. Dawson, N. W. Downes, F. E. Giesecke, J. M. Robertson, J. H. Kitchen, A. H. Sluss, G. L. Tuve.

This committee has been active during the year in analyzing the problem in order to develop a program which may give promise of satisfactory results. Through these efforts, a program involving the cooperation of Prof. F. E. Giesecke of the Agricultural & Mechanical College of Texas and Prof. E. F. Dawson of the University of Oklahoma has been developed.

This committee is charged in general with a study of the time-rate of heating buildings together with the extra heat load during the heating-up period in relation to the structure or type of building.

Technical Advisory Committee IP-7—SUMMER AIR CONDITIONING FOR RESIDENCES—M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts, Jr., Elliott Harrington, H. F. Hutzel, E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.

Work under this committee was continued at the University of Illinois in cooperation with the Laboratory during the past summer. A paper (Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F, by A. P. Kratz, S. Konzo and E. L. Broderick, A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 287), resulting from studies carried on during the summer of 1936, was presented to the Society at its Semi-Annual Meeting at Swampscott. A paper (Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz et al.), resulting from work carried on during the past summer, will be presented to the Society at the Annual Meeting in New York. The program for the past summer was as follows:

(1) The use of a materially undersized central plant mechanical condensing unit for cooling the first and second floors of the Residence with a unit having only 60 per cent of the capacity that was required to maintain 80 F on the most severe days.
(2) To investigate the problem of cooling the first floor during the day and the second floor during the night, using the small condensing unit.
(3) To obtain additional data on the length of the period between the time that the outdoor temperature reaches a maximum and the time that the heat conducted through the wall reaches a maximum.

Referring to item (2), it is thought desirable that some information on the rate at which the temperature on the second story can be reduced with several different cooling capacities should be obtained. This can be done easily with the condensing unit now installed in the Research Residence by merely changing the speed of the compressor.

The heat-lag and heat-flow studies made under item (3) during the past summer were with an uninsulated frame wall. It is believed that some interesting and useful data can be obtained on an insulated frame wall section by filling the studding spaces with insulation and repeating the test procedure during the summer of 1938.

Technical Advisory Committee IP-6—AIR FRICTION—J. H. Van Alsburg, *Chairman;* C. A. Booth, S. H. Downs, C. M. Humphreys, R. D. Madison, L. B. Miller, L. G. Miller.

Plans have been developed for a comprehensive study of the frictional resistance to the flow of air through ducts and fittings, ranging in size from the smallest used in small domestic air conditioning systems to those used in large central fan systems.

A set-up has been developed at the Laboratory in Pittsburgh and a number of preliminary observations made on an 8-in. round, an 8-in. square, and an 8-in. rectangular duct of a 4 to 1 aspect ratio. Considerable delay has resulted from failure to check the results of earlier investigators and the application of the Reynolds Number to the findings. However, these difficulties have been fairly well accounted for.

In the preliminary work considerable discrepancy or variations were found in the frictional resistance to the flow of air in factory-fabricated duct. These variations have lately been accounted for by a variable friction resulting from constrictions at the joints, which have been found to vary from the equivalent of from 2 to 6 ft

of length of duct. It has also been found that the frictional resistance for 8-in. duct is measurably lower than that in use.

Plans for the coming year include a study of factory fabricated and shop-made duct of several different sizes, ranging from 4 to 24 in.

Technical Advisory Committee IF-8—HEAT REQUIREMENTS OF BUILDINGS—O. W. Armspach, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, V. W. Hunter, H. H. Mather, E. C. Rack, F. B. Rowley, R. J. J. Tennant, J. H. Walker.

There has been little change in the comprehensive program developed by the committee over two years ago. The work necessary to follow out these plans is extensive and has engaged all of the research facilities both at Pittsburgh and elsewhere. During the year the Laboratory published another paper (Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet, A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 437), completing publication of the results of its study in the Grant Building in Pittsburgh.

Work is being continued at the University of Minnesota on the conductivity of concrete of various types. Professor Giesecke is continuing his studies of the heat requirements of some of the buildings at the Agricultural & Mechanical College of Texas, in cooperation with the Laboratory.

Among the additional activities of this committee is the analyzing of data collected at an earlier date relating to the heating requirements of Sibley Dome, one of the Cornell University Buildings. Other work of the committee has been a study of the degree-day method of estimating fuel consumption and a study of condensation and air return rates in the heating of a large school building at the University of Wisconsin.

While this committee has been quite active over a period of years, its current work has been lessened to a considerable degree by lack of finances with which to undertake specific projects. In the matter of cooling requirements for buildings, this phase of research was partially taken over during the year by the Committee on Cooling Load in Summer Air Conditioning, a brief description of whose work will be found in another section of this report. This involves a further study of heat transfer through roofs and walls of buildings as affected by heat capacity as well as solar radiation.

Technical Advisory Committee IF-18—AIR CONDITIONING REQUIREMENTS OF GLASS —M. L. Carr, *Chairman;* F. L. Bishop, A. N. Finn, E. H. Hobbie, R. J. Lillibridge, R. A. Miller, F. W. Parkinson, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker, G. B. Watkins, F. C. Weinert.

This committee, first appointed during 1936, has been particularly aggressive during the year. A sizable budget for the latter part of 1937 and throughout 1938 has been provided for, and a program including the following has been adopted for 1937-1938:

(1) Analysis, distribution to the committee and publication of available current data.
(2) Preparation and distribution to the members of the committee of a comprehensive bibliography.
(3) Cooperation with manufacturers and others carrying on research.
(4) Preparation of monthly reports of progress of the work carried on under the committee's budget for distribution to the committee.

Considerable progress under items (1) and (2) has already been made. At its last meeting a report of a study covering the requirements to heat two small houses, one with single and the other with double-glazed windows, was discussed by the committee with a view to preparing a technical paper. Like analyses of several other reports will also be made with a view to the publication of papers and it is the intention of the committee to issue reports on projects at monthly intervals throughout the coming year.

Technical Advisory Committee IF-23—INSULATION—L. A. Harding, *Chairman;* E. A. Allcut, H. C. Bates, H. C. Dickinson, J. D. Edwards, E. C. Lloyd, W. E. McMullen, R. T. Miller, E. R. Queer, T. S. Rogers, F. B. Rowley, W. S. Steele, C. Tasker, Dr. B. Townshend, G. B. Wilkes.

A comprehensive program of research activity was developed by this committee a year ago. This program included a review and a possible revision of the Society's Code for testing insulating materials and heat transmission through walls. During the past year the committee has been engaged in these Code revisions.

Because of the controversial nature of the subject, the work has proved more time-consuming than was anticipated.

Some score of subjects dealing with fundamental aspects of insulation constitute the general work of the committee. These include:

The effect of regain moisture on the conductivity of insulating materials.

The effect of condensed moisture on the conductivity of insulating materials in a structure.

Effect of the thickness of an insulating material accepted for test on the conductivity value obtained.

Effect of condensation on the surface of reflective insulation.

Effect of fibre size and fibre arrangement within an insulating board on its conductivity.

Relation between the heat capacity of an insulating material on the rate of heat penetration through a structure in which it is installed.

The degree of settling found for different fill materials after it is placed in the structure.

Technical Advisory Committee IF-1—SOUND CONTROL—J. S. Parkinson, *Chairman;* C. M. Ashley, G. F. Drake, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, A. E. Stacey, Jr., G. T. Stanton, F. R. Watson.

This committee has been active during the past few years in making available the results of research carried on by various members of the committee or others in the industry, and in correlating these results so as to form a sound basis for future research. A comprehensive program for future work has been drawn up, to be carried on as soon as the necessary budget provision has been taken care of.

A paper (The Nature of Noise in Ventilating Systems and Methods for Its Elimination, by J. S. Parkinson, A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 95), in the nature of a progress report, was presented to the Society at its St. Louis meeting.

Technical Advisory Committee IF-2—CORROSION IN STEAM SYSTEMS—A. R. Mumford, *Chairman;* W. H. Driscoll, C. A. Dunham, L. B. Miller, R. R. Seeber, F. N. Speller, C. M. Sterne.

The work of Professor Seeber at Michigan College of Mining and Technology, in cooperation with the Laboratory, was continued during 1937, under a program developed by the committee at an earlier date. Lack of funds somewhat reduced the activity during the latter part of the year.

The following program has been developed for continuation of the work during 1938:

(1) Analyses of condensate produced from steam of known composition in a tight heating system.
(2) Analyses of condensate produced from steam of known composition in an otherwise tight heating system fitted with commercial accessories.
(3) Extent and location of corrosion by oxygen-free condensate containing dissolved CO_2.
(4) Study of patented compounds claimed to be capable of stopping corrosion.
(5) Maintain a case history file of corrosion problems and their remedies.
(6) Cooperation with the *N.D.H.A.* in the revision of standard methods of sampling and analyzing steam and condensate.

Technical Advisory Committee C-11—PSYCHROMETRY—F. R. Bichowsky, *Chairman;* C. A. Bulkeley, J. A. Goff, Dr. E. Vernon Hill, F. G. Keyes, A. P. Kratz, W. M. Sawdon.

The committee has developed a program for first checking and then, if necessary, recalculating the psychrometric relationships going to make up the psychrometric chart.

It is apparent that certain theoretical errors exist in the data now available; whether or not these theoretical errors are of real practical importance can only be

determined by test calculations for a few widely distributed points on the psychrometric chart.

Technical Advisory Committee IP-26—ATMOSPHERIC IMPURITIES AND RESULTING SAFETY AND HEALTH REQUIREMENTS—Theodore Hatch, *Chairman;* J. J. Bloomfield, C. A. Booth, Philip Drinker, Dr. Leonard Greenburg, Elliott Harrington, H. B. Meller.

This committee has been active during the year correlating the work of various investigators interested in the subject.

The general objectives of the committee include:

(1) Collection and evaluation of existing published data pertaining to the industrial dust control problem.

(2) Outlining and guiding fundamental research problems by the Society.

(3) Fostering field studies for the collection of basic engineering data.

(4) Action to bring about the preparation and publication of more technical papers in this field. A great deal of valuable engineering work is being done in industrial establishments, but very little of this is described in technical journals in a form that is helpful to others.

Technical Advisory Committee IF-29—EFFECT OF WATER ON ROOFS—A. B. Snavely, *Chairman;* M. R. Beasley, J. B. Griffiths, Elliott Harrington, E. H. Hyde, W. L. Murray, E. R. Queer, C. S. Reeve, E. T. Selig, Jr.

This committee organized during this year and has been interested in a study of the insulating effect of water on roofs, either in the form of pools or in the form of a spray, on the heat transfer outward during the winter, or inward during the summer.

While water sprays on roofs have been fairly common, especially in the West and South, stationary water pools are a relatively newer development. Inasmuch as very little reliable information exists on the results that can be expected from water as an insulation medium on flat roofs, the committee was formed to study all phases of the subject. It is proposed among other things, to secure data in the way of accumulated temperature readings with dry roofs and later with water under similar circumstances.

Cooperating in Research

The following schools, colleges and other organizations cooperated during the year with the A.S.H.V.E. Committee on Research, and are continuing their investigations of the various problems listed:

Texas College of Agriculture & Mechanical Engineering—Hot Water Heating, Summer Comfort Cooling Requirements; University of Wisconsin—Aeration of Buildings, Effect of Entering Air on Temperature, Velocity and Distribution of Air in Enclosed Spaces; University of Minnesota—Air Cleaning Devices, Conductivity of Insulation and Building Materials, Heat Transfer Through Building Construction; University of Illinois (College of Engineering)—Direct and Indirect Radiation with Gravity Air Circulation, Residence Air Conditioning, (Medical School)—Air Conditioning in Treatment of Disease.

Case School of Applied Science—Heat Transfer of Finned Tubes with Forced Air Circulation; Michigan College of Mining and Technology—Corrosion in Heating Systems; Ontario Research Foundation—Summer Comfort Cooling Requirements; Princeton University—Measurement of Air Flow Through Ducts; University of Pittsburgh—Air Conditioning Requirements for Hospitals; Yale University (School of Public Health)—Treatment of Air with Electricity; Pittsburgh Testing Laboratories—Air Conditioning Requirements of Glass; Minneapolis-Honeywell Regulator Company—Summer Comfort Cooling Requirements.

Contributors to Research

Acknowledgment is made of financial contributions in support of the research work of the Committee on Research, from the following:

GENERAL RESEARCH

American Air Filter Co., Inc.; American Blower Corp.; Bayley Blower Co.; Barnes & Jones; Bell & Gossett Co.; Crane Co.; Clarage Fan Co.; C. A. Dunham Co.; Detroit Stoker Co.; Fairbanks, Morse & Co.; Julien P. Friez & Sons; Fulton Sylphon Co.; G. & O. Mfg. Co.; Gilbert & Barker Mfg. Co.; Grinnell Co.; *Heating, Piping and Air Conditioning Contractors National Association;* Hoffman Specialty Co.; Ilg Electric Ventilating Co.; Detroit Lubricator Co.; Young Radiator Co.; Independent Air Filter Co.; Illinois Testing Laboratories; Kewanee Boiler Co.; Kinetic Chemicals, Inc.; Mueller Brass Co.; May Oil Burner Corp.; Modine Mfg. Co.; Minneapolis-Honeywell Regulator Co.; Trane Co.; Timken Silent Automatic Div., The Timken-Detroit Axle Co.; Warren Webster Co.; York Ice Machinery Corp.; Independent Register Co.; J. P. Marsh Corp.; Economy Pump & Machine Co.; Electrol, Inc.; Auditorium Conditioning Corp.; Anemostat Corp. of America; Nash Engineering Co.

AIR CONDITIONING REQUIREMENTS OF GLASS

Owens-Illinois Glass Co.; American Window Glass Co.; Mississippi Glass Co.; Franklin Glass Corp.; Pittsburgh Plate Glass Co.; Blue Ridge Glass Corp.; Libby-Owens Ford Glass Co.; The U. S. W. Co.; Unique Balance Co., Inc.; Metal Window Institute; Pennsylvania Wire Glass Co.

RESIDENCE AIR CONDITIONING

Nash-Kelvinator Corp.

The following firms donated or loaned equipment to the Laboratory and for cooperative research projects:

American Instrument Co., Washington, D. C.; Buffalo Forge Co.; Dravo-Doyle Co., Pittsburgh; Julien P. Friez & Sons, Inc, Baltimore; Excelsior Steel Furnace Co.. Chicago; Illinois Testing Laboratories, Inc., Chicago; Kinetic Chemicals, Inc., Wilmington; Meyer Furnace Co., Peoria; Milcor Steel Co., Canton, Ohio; Minneapolis-Honeywell Regulator Co.; American Air Filter Co., Inc., Louisville; Anemostat Corporation of America, New York; Detroit Lubricator Co.; Acme Industries, Inc., Jackson, Mich.; Westinghouse Electric & Mfg. Co., Pittsburgh.

President Boyden called the fourth session, which was a joint session with the *American Society of Refrigerating Engineers,* to order and introduced the presiding chairman, W. H. Carrier, Syracuse, N. Y., past president of both societies. Before presenting the first technical paper of the session, Mr. Carrier introduced the president of the *A.S.R.E.,* H. M. Williams, Dayton, Ohio.

At the final technical session of the 44th Annual Meeting, L. J. Fowler, London, England, was introduced to the meeting and conveyed the special and best wishes of the president and council of the *Institution of Heating and Ventilating Engineers of Great Britain* to the Society. In closing, the speaker indicated how happy he was to come back and see so many friends that he made eight years ago when he was the author of an interesting paper entitled Panel Warming, which was presented before the Society.

The Report of the Committee on Code for Testing Stoker-Fired Steam-Heating Boilers was given by the chairman, C. E. Bronson, Kewanee, Ill.

Report of Committee on Code for Testing Stoker-Fired Steam-Heating Boilers

The Committee was assigned the duty of preparing a new Code for the Testing of Stoker-Fired Steam-Heating Boilers and of reviewing the present test codes to determine if revisions were essential.

In the Code for Testing Stoker-Fired Steam-Heating Boilers presented for your consideration, the general scheme of arrangement is similar to the Code for Testing Oil-Fired Boilers. Its provisions admit the burning of any type of solid fuel such as anthracite or bituminous coal. The type of stoker or boiler is not limited.

Particular attention is directed to the purpose of the Code, as expressed in the first sentence of the introduction. The Code is intended to provide a standard method

for conducting and reporting tests and to determine heat efficiency and performance characteristics. It does not fix the conditions that must be fulfilled for the acceptance or approval of the combination of boiler and stoker. This Code does not seek to specify the coal burning rate of a stoker required for the proper performance of a given boiler, but merely to report the results obtained with any combination.

Results of tests may be reported in the standard or long form or in the short-test form, except that for intermittent tests only the short-test form is required.

The Committee gave due consideration to reporting heat balance for intermittent tests, but on account of the difficulty in obtaining accurate average values throughout the test period, did not consider its inclusion advisable.

Furnace volume and hearth or grate area are defined for the purpose of this Code. No attempt has been made to define either effective furnace volume or active grate area. It is realized that combustion may be more intense in some particular zone, yet the proposed definitions outline practical values which can be used as a basis for comparison.

In reviewing the present test codes, it was the consensus of opinion that revisions are desirable. Both the general test code designated as Codes 1 and 2, also the Code for Testing Oil-Fired Steam-Heating Boilers, should be changed to eliminate reference to evaporation in pounds of water per pound of fuel, using instead the more widely used heat unit basis. The lack of time did not permit a more detailed or specific report on the changes required, and furthermore the committee feels that this duty should perhaps devolve upon a new committee whose personnel would recognize those primarily interested in oil burning.

Test Codes have an important bearing on the heating and ventilating industry and annual revision may be necessary to maintain them in up-to-date manner. It is hoped that a Standing Committee will be assigned for this duty.

It was regularly moved and seconded that the Code be presented to the Society membership for letter ballot.

The following amendments were read by the Secretary and voted on in accordance with Article B-XVI, and were adopted:

ARTICLE B-X—Local Chapters, *Section 1.* (*a*) Local Chapters of the Society may be formed upon application of twenty (20) members if the organization of such local Chapter will, in the judgment of the Council of the Society, advance the Society's interests.

ARTICLE B-XI—Funds, *Section 1.* When so directed by the Council, the Chairman of the Finance Committee shall invest such portion of any funds of the Society, as determined by the Council, in securities of the United States Government or securities legal for the investment of funds of savings banks of the State of New York.

Not less than one half of such invested funds shall be in securities of the United States Government. All investments shall be approved by the Council.

Installation of Officers

The newly elected officers, who were installed by Past President Homer Addams included Pres. E. Holt Gurney, Toronto, 1st Vice-Pres. J. F. McIntire, Detroit, 2nd Vice-Pres. F. E. Giesecke, College Station, Tex., and Treas. A. J. Offner, New York. The four new members of the Council, N. D. Adams, Rochester, Minn., J. H. Walker, Detroit, Prof. A. P. Kratz, Urbana, Ill., and G. L. Wiggs, Montreal, were escorted to the rostrum and introduced.

Resolutions

On motion of J. H. Van Alsburg, chairman of the Resolutions Committee, seconded by T. H. Urdahl, it was voted that this expression of thanks and appreciation be adopted by the members attending the 44th Annual Meeting:

We have had a very wonderful Annual Meeting, which is only partially proven by the registration. This is one of the largest meetings we have ever had and total registration exceeds 1000. We are having the best and largest Exposition and we

have many to thank for the fine work they have done in making this meeting such a wonderful success.

We wish to definitely record our appreciation of the efforts of those who individually and collectively have made this meeting outstanding in Society history.

We, first of all, thank the New York Chapter and its Committee on Arrangements for the very fine work they have done in providing the social features which contributed so materially in making this meeting a success. Especially, commending them for the wonderful entertainment provided for the ladies.

We thank the management of the Hotel Biltmore for the many courtesies and exceptional services which they have rendered in providing a maximum of comfort during our stay.

We thank the press and trade publications for the fine publicity they have given the Society.

We thank the authors of papers who have contributed so largely to the interest in attendance at our technical sessions.

We express our sincerest appreciation to Mr. John B. Kennedy for his inspiring address at our 44th Annual Banquet and thank Mr. Jack Knight for his enjoyable and educational talk.

We thank the exposition management and the advisory committee for their efforts in making this Exposition the largest and best in the history of the Society.

We thank the *American Society of Refrigerating Engineers* and the *National Warm Air Heating and Air Conditioning Association* for their cooperative effort in bringing our respective organizations, our members and our ideals in the art of heating, ventilating and air conditioning into more intimate contact.

Finally and with particualr emphasis, we thank our good friend and Secretary, A. V. Hutchinson, and his very capable staff whose material effort underlies the execution of all work of the Society and whose fine spirit is so largely responsible for the familiar attitude of our members toward one another. The inadequacy of words alone makes it difficult to properly express the feeling in the hearts of the membership on whose behalf this resolution is gratefully submitted.

After calling for new business, to which there was no response, the presiding chairman, Vice-President McIntire, adjourned the 44th Annual Meeting of the Society at 12:05 p. m.

PROGRAM, 44th ANNUAL MEETING, 1938

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL BILTMORE, NEW YORK, N. Y.
JANUARY 24-28, 1938

Sunday, January 23

10:00 A.M. Research Committee on Air Conditioning Requirements of Glass
12:30 P.M. Council Luncheon and Meeting

Monday, January 24

9:30 A.M. Registration—Fountain Court (19th Floor)
10:00 A.M. Conference of Chapter Delegates
Meeting of Committee on Research
12:30 P.M. Council Meeting
2:00 P.M. Opening of Heating and Ventilating Exposition
Meeting of Nominating Committee
Meeting of Fan Test Code Committee
6:30 P.M. Get-Acquainted Dinner (Bowman Room)
President's Reception

Tuesday, January 25

9:00 A.M. Registration—Fountain Court (19th Floor)
9:30 A.M. First Session—Pres. D. S. Boyden presiding
Greetings—W. E. Heibel, President, New York Chapter

Response—Pres. D. S. Boyden
Reports of Committees
Technical Papers:
Application and Economy of Steam Jet Refrigeration to Air Conditioning, by A. R. Mumford and A. A. Markson
Cooling Tower Equipment and Its Relation to Water Conservation, by S. I. Rottmayer
Report of Tellers of Election

11:00 A.M. Research Committee on Air Friction
Research Committee on Insulation

12:00 NOON International Heating and Ventilating Exposition—Grand Central Palace

2:00 P.M. Joint Session with *National Warm Air Heating and Air Conditioning Association*—E. Holt Gurney, 1st Vice-Pres., presiding
Technical Papers:
Fundamentals Developed from Twenty Years of Research by the National Warm Air Heating and Air Conditioning Association, by A. P. Kratz
Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart
Condensation within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund

Wednesday, February 26

8:30 A.M. Research Committee on Intermittent Heating

9:30 A.M. Third Session—Pres. D. S. Boyden presiding
Technical Papers:
Studies on Bacterial Control in Air Conditioning, by T. S. Carswell, J. D. Fleming and H. K. Nason
Physiologic Response of Man to Environmental Temperature, by Dr. F. K. Hick, Dr. R. W. Keeton and Nathaniel Glickman
Control of Body Heat Loss Through Radiant Means, by Dr. C. A. Mills and Dr. Cordelia Ogle
Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley

12:00 NOON International Heating and Ventilating Exposition—Grand Central Palace
Research Committee on Air Distribution

2:00 P.M. Joint Session with *American Society of Refrigerating Engineers*—W. H. Carrier, Past President *A.S.H.V.E.* and *A.S.R.E.*, presiding
Technical Papers:
Control of Air Conditioning in Large, Medium, and Small Buildings, by W. E. Zieber and S. F. Nichol
Physiological Reactions and Sensations of Pleasantness under Varying Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge
Performance Tests of Asbestos Insulating Air Duct, by R. H. Heilman and R. A. MacArthur

6:30 P.M. Past Presidents' Dinner

Thursday, January 27

9:30 A.M. Fifth Session—Pres. D. S. Boyden presiding
Technical Papers:
Effects of Artificial Lighting on Air Conditioning, by Walter Sturrock
Comparative Analysis of Office Building Air Conditioning Systems, by J. R. Hertzler
Heating and Ventilating the Lincoln Vehicular Tunnel, by C. W. Murdock

12:00 NOON International Heating and Ventilating Exposition—Grand Central Palace

2:00 P.M. Research Committee on Corrosion in Steam Heating Systems

4:00 P.M. Research Committee on Corrosion in Air Conditioning Equipment

7:00 P.M. Annual Banquet—Presentation of Past President's Emblem to D. S. Boyden

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Friday, January 28

9:30 A.M. Sixth Session—Pres. D. S. Boyden presiding
Technical Papers:
Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth, by F. C. Houghten, Carl Gutberlet and Edward Witkowski
Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo
Heat Transfer through Single and Double Glazing, by M. L. Carr, R. A. Miller, A. C. Byers and Leighton Orr
Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley
12:00 NOON Council Meeting
International Heating and Ventilating Exposition—Grand Central Palace
2:00 P.M. Meeting of Guide Publication Committee

ENTERTAINMENT EVENTS

Monday, January 24

6:30 P.M. Get-Acquainted Dinner, President's Reception—Bowman Room

Tuesday, January 25

10:30 A.M. Ladies Shopping Tour
2:30 P.M. Ladies Bridge
10:00 P.M. International Casino Party

Wednesday, January 26

10:30 A.M. Ladies Tour of Radio City and NBC Studios
2:00 P.M. Ladies Theater Party
6:30 P.M. Past Presidents' Dinner
8:00 P.M. Adventures in Science, by Dr. Phillips Thomas—Electrical Association Auditorium, Grand Central Palace

Thursday, January 27

10:30 A.M. Trip of Inspection, S.S. Rex, for Ladies
12:45 P.M. Ladies Theatre Party, Radio City Music Hall
2:00 P.M. Inspection Trip, Lincoln Vehicular Tunnel
Inspection Trip, Beechnut Plant
7:00 P.M. Annual Banquet—Ballroom. *Speaker*—John B. Kennedy; *Toastmaster*—W. H. Driscoll. Presentation of Past President's Emblem to D. S. Boyden

COMMITTEE ON ARRANGEMENTS

A. C. BUENSOD, *General Chairman* ALFRED ENGLE, *Vice-Chairman*

Inspection and Transportation: A. J. Offner, *Chairman;* H. L. Alt, A. L. Baum, J. G. Eadie, C. S. Koehler, T. W. Reynolds, G. W. Martin.

Entertainment: G. E. Olsen, *Chairman;* Louis Hament, A. F. Hinrichsen, E. J. Ritchie, W. J. Osborn, K. L. Mytinger.

Ladies: H. W. Fiedler, *Chairman;* Mrs. H. W. Fiedler, W. H. Ballman, Mr. and Mrs. D. J. Purinton, R. C. Bonthron, Henry Gitterman, F. B. Herty, W. A. Sherbrooke.

Banquet: Russell Donnelly, *Chairman;* A. A. Adler, V. J. Cucci, R. W. Cumming, H. B. Hedges, C. R. Hiers, H. G. Meinke, C. H. Quirk, B. H. Schulze.

Finance: F. E. W. Beebe, *Chairman;* W. M. Heebner.

Reception: W. W. Timmis, *Chairman;* L. W. Charlet, M. E. Durkee, J. R. Murphy, R. D. Tyler, N. P. Fenner, H. P. Waechter, L. L. Munier.

Publicity: C. H. B. Hotchkiss, *Chairman;* R. V. Sawhill, John James.

No. 1073

CODE OF MINIMUM REQUIREMENTS FOR COMFORT AIR CONDITIONING*†

SECTION I—PURPOSE, SCOPE AND LIMITATION

THE purpose of this code is to provide a uniform procedure and methods for use in establishing the fundamental basis for the design of comfort air conditioning installations in the United States and Canada. The requirements herein set forth are intended to serve as minimum design requirements, and are not to be construed as limiting good practice nor preventing progress in the art of air conditioning. These requirements are intended to be applied only to systems provided with mechanical circulation of the air.

SECTION II—DEFINITIONS AND ABBREVIATIONS

1. **Comfort Air Conditioning,** for the purpose of this code, is defined as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all of these functions, it shall be designated by a name that describes only the function or functions performed.

2. **Enclosure,** for the purpose of this code is defined as the space to be served by comfort air conditioning apparatus.

3. **Design Load,** for the purpose of this code is defined as the estimated capacity of the apparatus required to provide and maintain specified inside conditions with specified conditions of temperature and humidity existing outside when the summation of all sources of load that occur coincidentally is a maximum.

4. **Effective Temperature** as defined in THE A.S.H.V.E. GUIDE is an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity and movement of the air.

5. A.S.H.V.E.—AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.
A.S.R.E.—American Society of Refrigerating Engineers.
A.S.A.—American Standards Association.

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* Sponsored by Joint Committee of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers.*

COMMITTEE: L. A. HARDING, *Chairman;* GLENN MUFFLY, *Vice-Chairman;* A.S.H.V.E.: D. E. FRENCH, JOHN HOWATT, A. P. KRATZ, C. W. WALTON, JR.; *A.S.R.E.:* W. L. FLEISHER, L. A. PHILIPP, H. J. MACINTIRE, W. E. ZIEBER.

† Adopted by AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, January 25, 1938.

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SECTION III—DESIGN TEMPERATURES AND RELATIVE HUMIDITIES

1. **Design Outside Air Temperature and Relative Humidity Conditions** are construed to mean the dry- and wet-bulb temperatures (or relative humidities) specified for design load computations.

2. The design outside dry-bulb temperature in various cities for use in computing the design heating load shall be obtained from the latest edition of THE A.S.H.V.E. GUIDE.

3. The design outside dry- and wet-bulb temperatures in various cities for use in computing the design cooling load shall be obtained from the latest edition of THE A.S.H.V.E. GUIDE.

4. **Design Inside Air Temperature and Relative Humidity Conditions** are construed to mean the dry- and wet-bulb temperatures (or relative humidities) specified to be maintained inside of the enclosure at the time of the occurrence of the design load.

5. The design inside dry-bulb temperature for the design heating load shall be assumed at 70 F.

6. The design inside relative humidity for the design heating load shall be assumed as 35 per cent with an assumed outside dry-bulb temperature 30 F.

7. The design inside dry-bulb temperature and relative humidity for the design cooling load, corresponding to the outside dry- and wet-bulb temperature, Section III—paragraph 3, shall be based on an effective temperature selected from the following table, for occupancies in excess of 40 min and for sedentary occupations; between relative humidity limits of 30 and 75 per cent inclusive and the air velocity limits as hereinafter stated in Section VI—paragraph 2.

DESIGN OUTSIDE DRY-BULB TEMP. DEG F	DESIGN INSIDE AIR CONDITIONS EFFECTIVE TEMP.
105	75.5
100	75
95	74
90	73
85	72
80	71

SECTION IV—DESIGN LOAD ESTIMATES

1. **Design Heating Load.** The rate at which heat is assumed to be supplied the enclosure shall be the sum of the individual rates as determined by the following items.

2. Heat transfer through the windows, walls, partitions, doors, floors, skylights, ceiling or roof of the enclosure due to the air dry-bulb temperature difference assumed to exist between the air on the opposite sides of the construction considered.

3. Heat to increase the dry-bulb temperature of the outside air and its contained superheated vapor assumed to enter the enclosure by infiltration or purposely intro-

duced for ventilation, whichever is the greater, to the inside design dry-bulb air temperature.

4. Heat to evaporate the necessary water required and superheat the resulting vapor in order to raise the moisture content of the outside air assumed to enter the enclosure by infiltration or purposely introduced for ventilation, to the inside design relative humidity.

5. It shall be clearly stated in both the specifications and proposal if heat credit is to be allowed or permitted for any heat emitting source located within the enclosure.

6. **Design Cooling Load.** The rate which heat is assumed to be extracted from the enclosure to compensate for the sum of the individual rates as determined by the following items.

7. Heat transfer through the windows, walls, partitions, doors, floors, skylights, ceiling or roof of the enclosure due to the air dry-bulb temperature difference assumed to exist between the air on the opposite sides of the construction considered.

8. Sensible heat emission of the occupants within the enclosure. The assumed number of occupants within the enclosure shall be the stated number as designated by the owner as estimated during the time when the sum of the remaining estimated loads is a maximum.

9. Heat emission of electrical, chemical, gas, steam, hot water or other devices, apparatus or lights located within the enclosure.

10. Transfer of solar radiation through the windows, walls, doors, skylights or roof of the enclosure.

11. Heat to be extracted to lower the dry-bulb temperature of the air and its contained super-heated vapor (commonly termed moisture content) assumed to enter the enclosure from the outside to the design inside temperature condition.

12. Heat to be extracted in order to remove the vapor entering the enclosure with the assumed outside entering air including vapor emitted by persons and any other vapor emitting sources located within the enclosure in order to maintain the inside design relative humidity.

SECTION V—SOURCE FOR HEAT TRANSFER, INFILTRATION AND VENTILATION RATES

1. **Heat Transfer Rates for Building Construction.** The unit heat transmission rates for windows, walls, partitions, doors, floors, skylights, ceiling or roof of the enclosure to be employed shall be the rates, for the types of construction involved, obtained from the latest edition of THE A.S.H.V.E. GUIDE, or as may be calculated by employing the conductivity and combined rates of radiation and convection given by this publication.

2. **Heat and Moisture Emitted by Persons.** The estimated rate of heat and moisture emitted by the occupants in the enclosure, for the class of activity involved, shall be obtained from the latest edition of THE A.S.H.V.E. GUIDE.

3. **Heat Emission from Appliances.** The estimated rate of heat emitted by various mechanical, chemical, steam, hot water, gas and electrical appliances located within the enclosure shall be obtained from the latest edition of THE A.S.H.V.E. GUIDE. If any such appliance is located beneath an exhaust hood, served by an exhaust fan, only one-half of the amount stated need be considered. Other known sources of heat within the enclosure such as steam and hot water pipes, etc., shall be included.

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4. **Solar Heat Entering Enclosure.** The rate at which solar heat is assumed to enter the enclosure shall be obtained from the latest edition of THE A.S.H.V.E. GUIDE.

5. **Infiltration.** The rates at which air is assumed to enter the enclosure by window and door leakage shall be those for the types of windows and doors involved, obtained from the latest edition of THE A.S.H.V.E. GUIDE. If the rate at which air is specified to be introduced or removed from the enclosure by positive means exceeds the estimated infiltration rate use the greater rate.

6. **Ventilation.** The assumed rate at which outside air is to be positively introduced into the enclosure per occupant, when the contamination of the air within the enclosure results entirely from respiratory processes shall be not less than 10 cfm per stated number of occupants. (See Section IV—par. 8.)

The assumed ventilation rates shall be not less than 15 cfm per stated number of occupants in enclosures in which smoking is customarily permitted.

Provision shall be made for air removal from the enclosure either by natural or mechanical means at not less than the assumed ventilation rate.

7. **Air Quality or Purity.** For the purpose of this code the requirements of air quality or purity are assumed to be met if means are provided for the positive introduction of outside air in the amounts set forth in Section V, paragraph 6, and for removal of 95 per cent by count of all dust particles over 10 microns in diameter from all air delivered to the enclosure.

SECTION VI—AIR DISTRIBUTION

1. The quantity, air motion and temperature of the treated air and the method of introducing it to the conditioned space shall be designed to limit the variation in dry-bulb temperature to 3 F or less at a 5 ft level throughout that portion of a given room that is normally frequented by persons. The temperature shall not be taken at a point close enough to a wall or source of heat to be appreciably affected by it.

2. It is desirable to avoid air velocities exceeding 50 linear feet per minute in the zone between the floor and the 5 ft level, in spaces normally frequented by persons who are not normally moving about.

Exception may be made in the vicinity of a supply or return grille when construction necessity requires the grille to be located below the 5 ft level and in a space normally frequented by occupants.

3. It is desirable, for cooling, that the difference between the temperature of air currents in the space frequented by occupants and the average temperature in such space be not greater than as shown below for the various air current velocities. Velocity and temperature measurements should be made 5 ft above the floor level.

AIR CURRENT VELOCITY LINEAR FEET PER MINUTE	TEMPERATURE DIFFERENCE DEG F AVG ROOM TEMP. MINUS AIR CURRENT TEMP.
40 and over	2
Less than 40	3

4. **Requirements for Duct Capacity.** It is desirable that return ducts shall be of sufficient area to convey 100 per cent of the air handled by the fan to permit economical heating and cooling prior to occupancy.

It is desirable, to permit economical operation during mild weather, that the outside air duct be of sufficient area to convey approximately the total quantity of the air handled by the fan and that means be provided for the escape of this air quantity. The outside air duct shall in every case be of sufficient area to permit the minimum ventilation requirements to be met.

SECTION VII—APPARATUS RATING

Fans employed in connection with air conditioning systems shall be rated in accordance with the provision of the A.S.H.V.E. STANDARD TEST CODE FOR DISC AND PROPELLER FANS, CENTRIFUGAL FANS AND BLOWERS. Air heaters are to be rated in accordance with the provisions of the A.S.H.V.E. STANDARD CODE FOR TESTING AND RATING STEAM UNIT HEATERS. Unit ventilators shall be rated in accordance with the provisions of the A.S.H.V.E. STANDARD CODE FOR TESTING AND RATING STEAM UNIT VENTILATORS. Air cleaning devices shall be rated in accordance with the provisions of the A.S.H.V.E. STANDARD CODE FOR TESTING AND RATING AIR CLEANING DEVICES USED IN GENERAL VENTILATION WORK. Comfort cooling apparatus and refrigerating machinery are to be rated in accordance with *A.S.R.E.* Standards, which include *Standard Method of Rating and Testing Air Conditioning Equipment and Standard Method of Rating and Testing Mechanical Condensing Units.*

SECTION VIII—REQUIREMENTS FOR THE SELECTION OF REFRIGERATING EQUIPMENT

1. City water temperature selected shall not be less than the maximum values obtained from the latest edition of THE A.S.H.V.E. GUIDE.

2. Well water temperature selected shall not be less than the maximum value for the months of July and August for the locality.

3. A wet-bulb temperature shall not be assumed less than the outside design wet-bulb for the rating and selection of cooling towers and evaporative condensers.

4. A dry-bulb temperature shall not be assumed less than the outside design dry-bulb temperature for the rating and selection of air cooled condensers.

SECTION IX—NOISE CONTROL

1. Noise measurements for the enclosure shall be made at a height of 5 ft from the floor. All readings shall be taken 5 ft directly in front of any register face or air conditioning equipment or directly underneath when the register face or air conditioning equipment is located on the ceiling. The normal noise level at each of these locations shall be assumed to be the numerical average of five readings.

2. That portion of the noise resulting from the operation of an air conditioning system shall not be more than as stated in the specifications above the loudness level for the different locations. The enclosure loudness levels during normal occupancy, without the air conditioning system in operation, shall be as stated for average noise levels as given in the latest edition of THE A.S.H.V.E. GUIDE unless otherwise specified.

3. The loudness level above the noise in the enclosure and the noise produced by the air conditioning system shall be expressed in decibels based on the 40 decibel response curve and measure with a sound level meter constructed according to *A.S.A. Tentative Standards for Sound Level Meters* Z24.3-1936.

SECTION X—GUARANTEES AND EXCEPTIONS

1. **Guarantees** of performance shall be limited to stated performance and capacities that are described and can be measured, and the guarantees shall state the condition under which such capacities or results are to obtain.

2. **Exceptions.** If a proposed installation differs from the provisions in this code, the particular in which the design departs from the code shall be stated in both the specifications and proposal.

No. 1074

APPLICATION AND ECONOMY OF STEAM JET REFRIGERATION TO AIR CONDITIONING

By A. R. MUMFORD * AND A. A. MARKSON ** (*NON-MEMBERS*), NEW YORK, N. Y.

WITH the introduction of steam jet refrigeration, district heating companies became interested in it as a potential source of summer load and improvement in annual load factor.

On Fig. 1 are plotted the monthly total sales of steam to consumers of the New York Steam Corp. for the calendar year 1936. Two dotted sections have been added to the curve to indicate the increase in summer load if 10 per cent or 20 per cent of the space now served with heating steam should also be cooled by steam jet refrigeration. The ratio of the average month to the peak month is shown below each curve as a percentage. Although this is not the usual load factor, it is a relative figure and indicates a possible improvement which ought to be mutually beneficial. As in every new development, the particular conditions for which steam jet had to be designed were not fully understood in the beginning. The first machines were sold on the basis of first cost and simplicity. The tendency was to model the installations after the well established features of compression machines. There are 18 machines now operating on the utility service in New York City, aggregating 1560 tons in capacity. Most of these were satisfactory as installed, but in some cases the utility had to work on some of them with the cooperation of the manufacturers. The results of this effort have been to help establish this type of refrigeration on a sound basis. In order to be able to assist those of its consumers who had installed such equipment or who were interested in it, a flexible program of tests and investigations was introduced to provide the basic information for sales and service in cooperation with the manufacturers and consulting engineers.

As compared with other refrigerants being used for air cooling, water vapor may be said to come into its own, thermodynamically, above temperatures of 40 F. Air cooling consists of removing heat from an air-water vapor mixture. This heat, in the form of latent and sensible, is then pumped by a heat pump to the final heat level. In doing this practically, it is important to realize that the heat may have been transformed several times from one form to another merely for the convenience of the particular method. The steam jet machine

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as applied to air cooling may be considered as one in which a certain amount of space heat, sensible and latent, is first converted into the same quantity of heat entirely sensible, which heat is then pumped to the condenser in the form of water vapor or latent heat.

Steam jet cooling may therefore be regarded as a process of air cooling in which refrigeration is done on water vapor and air by means of water vapor and air without the use of special refrigerants to act as heat carriers. Below certain cooling temperatures a refrigerant of better thermodynamic properties than water vapor may be required. Thus, one of the main points of this paper is the discussion and demonstration of the air conditioning appli-

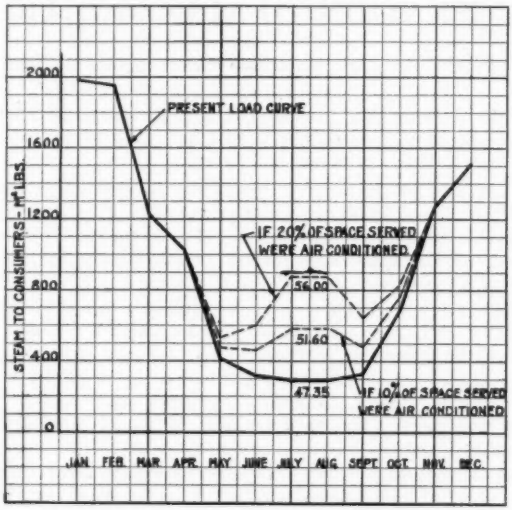

FIG. 1. ESTIMATED EFFECT OF SUMMER LOAD WHICH MIGHT BE ADDED BY STEAM JET REFRIGERATION

cations which fall into the economical range of water vapor refrigeration. It is believed that consideration of these demonstrations will inform engineers familiar with steam jet that objections to steam jet, based on the specifications of very low water temperatures, arise only when its operation is improperly compared with that of older systems.

Secondly, it is the purpose of this paper to demonstrate that steam jet as a direct process should be treated as such in its application in order to obtain real operating advantages and economies.

Third, a thermodynamic analysis is presented of the jet itself to familiarize engineers with the load characteristics of these machines obtainable by reduced steam pressure operation at partial cooling tower loads and the controls by which these are obtained.

Last, a discussion is given of the really fundamental practical matter of condenser water from a constructive engineering viewpoint.

SELECTION OF DEW-POINT OR FLASH TANK TEMPERATURE

The selection of the proper refrigeration temperature depends on so many conditions that a discussion of them all is beyond the scope of this paper. However, the factors which fix the proper dew-point can be discussed for two conditions, namely with and without the use of sensible reheat.

Without reheat, the dew-point is fixed by the ratio of sensible heat picked up in the conditioned space to total heat pickup. On Fig. 2 is shown a curve of dew-points which correspond to the heat ratios picked up in the conditioned

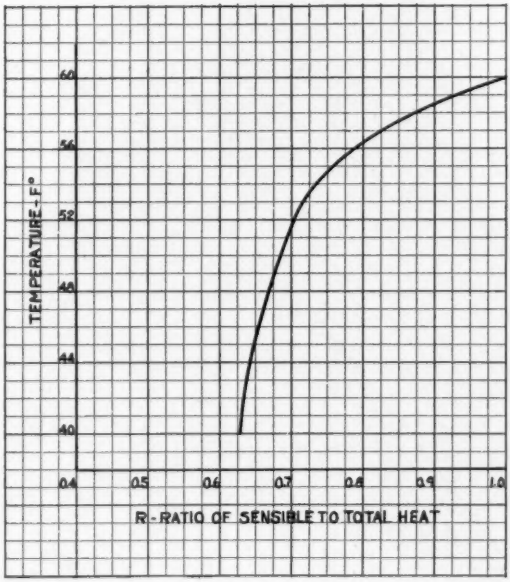

FIG. 2. VARIATION OF DEW-POINT WITH RATIO OF SENSIBLE TO TOTAL HEAT PICKED UP

space when the space is to be maintained at 80 F dry-bulb and 50 per cent relative humidity. For ratios much below 0.63 sensible to total heat pickup sensible reheat should be used. The limiting dew-point as shown is 60 F which is the highest allowable for 100 per cent sensible heat pickup under the maintained conditions of 80 F dry-bulb and 50 per cent relative humidity. As more and more latent heat is picked up the maximum permissible dew-point drops until a practical minimum of 40 F is reached at the 63 per cent ratio.

If the ratio of sensible heat to total pickup in a conditioned space is considered as the determinant of the character of the load to distinguish it from the size of the load, it can be stated that the character of the load determines the only possible dew-point which can be used without reheat to meet any fixed condition to be maintained in the space. To illustrate the relation of

dew-point to load characteristic, a series of psychrometric charts [1] are presented which are peculiarly well suited for the purpose.

If on a standard psychrometric chart plotted with moisture content per pound of dry air against dry-bulb temperature, a line be drawn between a point in the diagram and a point on the saturation line, it is seen that the

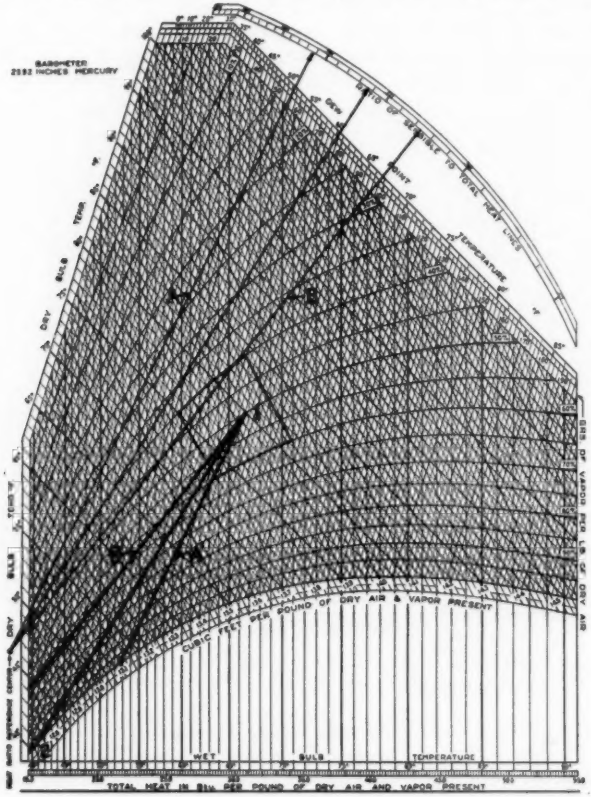
BAROMETER
29.92 INCHES MERCURY
RATIO OF SENSIBLE TO TOTAL HEAT LINES
DEW POINT TEMPERATURE
DRY BULB TEMPERATURE
GRAINS OF MOISTURE PER LB OF DRY AIR
CUBIC FEET PER POUND OF DRY AIR & VAPOR PRESENT
WET BULB TEMPERATURE
TOTAL HEAT IN B.t.u. PER POUND OF DRY AIR AND VAPOR PRESENT

FIG. 3. A FIXED CONDITION WITH FIXED PICK UP RATIO CAN BE MET BY ONLY ONE DEW-POINT WITHOUT REHEAT †

ratio of latent to sensible heat is practically constant along this line. The chart illustrated in this paper is a specialization of this principle in which the psychrometric chart has been erected on a framework of such ratio lines (sensible to total heat) yielding a clear method of analyzing practical psychrometric problems. As applied to these problems, this chart has an appeal similar to that of the Mollier steam diagram in steam power problems.

[1] The Psychrograph, by A. M. Norris (*A.S.M.E. Transactions*, July, 1935).

† Charts (Figs. 3 to 8) reproduced by permission of the Westinghouse Electric and Mfg. Co.

On Fig. 3 is indicated a fixed condition (Point 1) in the comfort zone at 80 F dry-bulb and 50 per cent relative humidity which it is desired to maintain within the conditioned space. The washer dew-point (Point 2) was selected arbitrarily at 40 F and the line connecting the fixed condition with the dew-point is the locus of all points of mixtures of air, recirculated and conditioned,

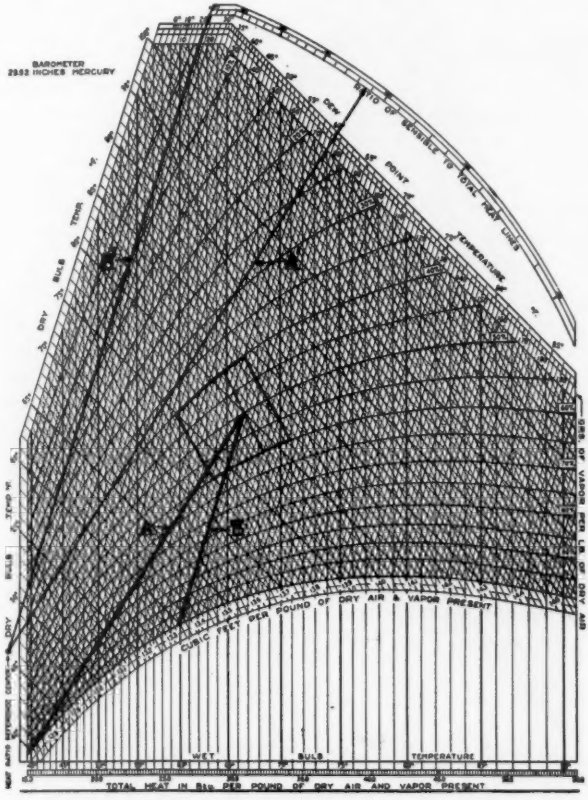
BAROMETER
29.92 INCHES MERCURY
LIMIT OF SENSIBLE TO TOTAL HEAT LINES
DEW POINT TEMPERATURE
DRY BULB TEMPERATURE
GRS OF VAPOR PER LB OF DRY AIR
CUBIC FEET PER POUND OF DRY AIR & VAPOR PRESENT
WET BULB TEMPERATURE
TOTAL HEAT IN Btu PER POUND OF DRY AIR AND VAPOR PRESENT

FIG. 4. RANGE OF PICK UP RATIOS WHICH CAN BE MET BY VARYING THE DEW-POINT WITHOUT REHEAT †

which will satisfy the desired conditions without reheat. On this type of chart, if a line is drawn through the reference center intersecting the arc of ratios at the upper part of the chart, and parallel to the line 1—2, the point of intersection is the unique ratio of sensible to total heat pickup in the conditioned space which will permit the desired condition to be maintained with a 40 F dew-point and without reheat. If it is assumed, for instance, that a new

† Loc. Cit., see p. 36.

reference center line, A, is drawn, showing an increase in sensible heat pickup in the conditioned space, it is obvious that its parallel, A', through the desired condition will intersect the saturation line at a higher temperature and will require a higher dew-point to satisfy the conditions. Similarly, if line B is

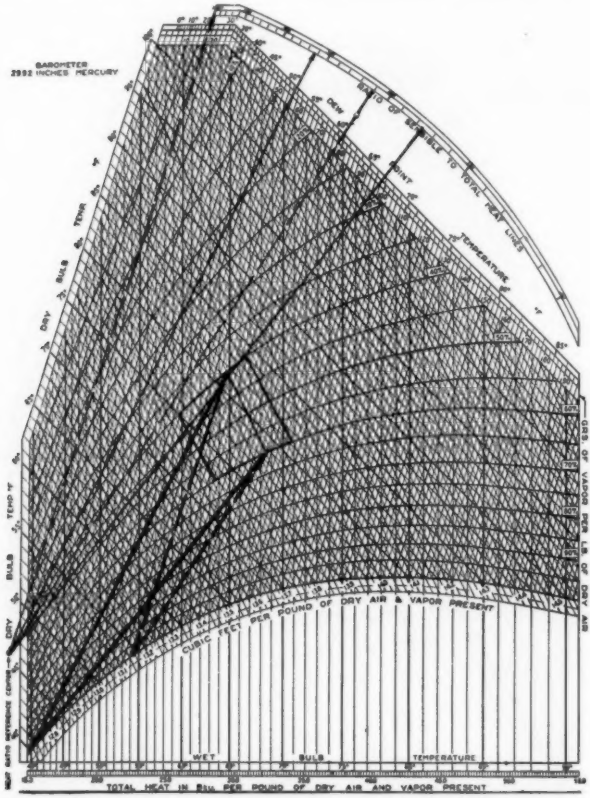
BAROMETER
29.92 INCHES MERCURY
DRY BULB TEMP °F
WET BULB TEMP °F
TEMPERATURE
TOTAL HEAT IN BTU PER POUND OF DRY AIR AND VAPOR PRESENT
GRS OF VAPOR PER LB OF DRY AIR
CUBIC FEET PER POUND OF DRY AIR & VAPOR PRESENT

FIG. 5. RANGE OF PICK UP RATIOS WHICH CAN BE MET BY LOW DEW-POINT AND BY HIGH DEW-POINT WITHOUT REHEAT IF CONDITIONS ARE ALLOWED TO VARY ALONG EFFECTIVE TEMPERATURE LINE †

drawn through the reference center corresponding to an increase in latent heat pickup in the conditioned space, its parallel, B', will intersect the saturation line at some dew-point temperature below 40 F or not at all. Thus, it can be stated that for a fixed condition to be maintained with a fixed dew-point only one character of load may be met. Conversely, for a fixed condition,

† Loc. Cit., see p. 36.

the dew-point must be varied to correspond with changes in the character of the load.

To examine this relation of dew-point to load character further, the original assumptions were changed to permit variation in the dew-point. On Fig. 4 is shown the same fixed condition to be maintained in the conditioned space without reheat. The lines *A* and *B* from the fixed condition to the saturation line represent the limiting lines of mixtures. The line *A* is drawn to the practical lowest dew-point of 40 F and the line *B* to the highest possible dew-point at about 60 F. The reference center parallels, *A'* and *B'*, indicate that the character of the load may change from a ratio of 100 per cent sensible to total heat pickup to one of about 63 per cent. Within this range of load character, it is possible, therefore, to maintain the desired conditions in the space exactly by raising the dew-point as the proportion of sensible heat pickup increases.

Not only is the advantage of maintaining exact conditions obtained but it is possible to operate at increasing economy as the dew-point rises because of the flash tank temperature—steam consumption characteristic of the steam jet machine.

If assumptions of fixed conditions are replaced with reasonable allowances for variations, possible means are developed for improving operating economy. The assumptions for Fig. 5 are that conditions be maintained in the space at any point along the effective temperature line, that no reheat is to be used, but that any suitable dew-point may be selected for the operation under consideration. Conditions have been assumed to be maintained on an effective temperature line rather than that they be allowed to stray throughout the comfort zone under the impression that wide variations, though within the comfort zone, may not result in complete satisfaction. Dew-points of 40 and 55 F have been selected as representing low and high dew-point operation. Limiting lines of mixtures have been drawn to the dew-points from the ends of the effective temperature line. The corresponding parallels have been drawn from the reference center to the ratio arc and these intersections indicate the ranges of load character which may occur in the conditioned space and yet have the maintained condition remain somewhere along the effective temperature line. Both the position of the range and the size of the range are different for the two dew-points. The higher dew-point, as would be expected, cannot handle as great a latent heat load without reheat as the lower dew-point. Conversely, the lower dew-point cannot handle as high a proportion of sensible heat as the higher dew-point.

The approximate changes in the character of the load which may take place in theaters, restaurants, office buildings and stores are shown in Table 1.

TABLE 1—ESTIMATE OF RANGE OF CHANGE OF LOAD CHARACTER WHICH INSTALLATIONS SHOULD BE EXPECTED TO HANDLE

TYPE OF OCCUPANCY	RATIO OF SENSIBLE HEAT TO TOTAL HEAT PICKED UP
Ballroom, Night Club, etc	54–80
Theater Building	70–90
Store	74–90
Office	78–90

One of the important variables is the person load which, in most cases, is the principal source of latent heat load. It is obvious that this load will vary during the day and, if comfort conditions are to be carefully maintained without reheat, the dew-point must be changed to suit the new ratio. It is to be noted that control along the effective temperature line within the comfort zone

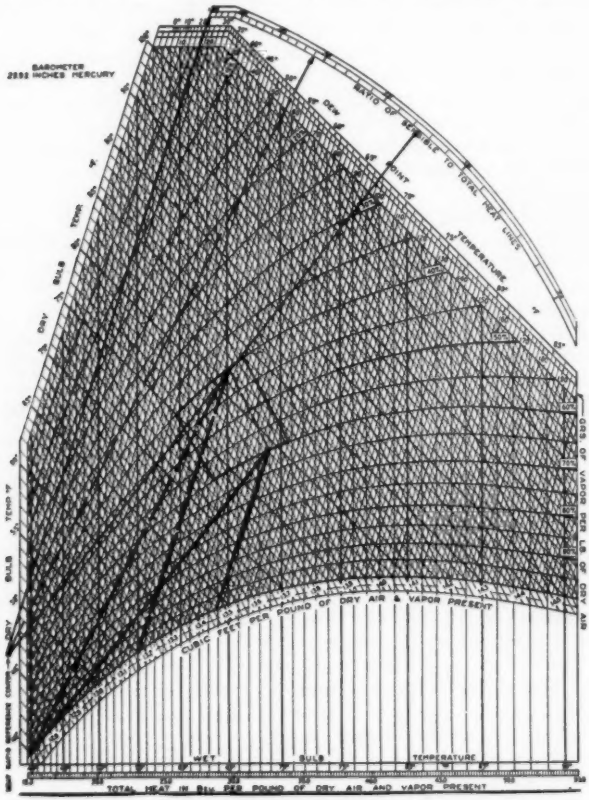
BAROMETER
29.92 INCHES MERCURY
RATIO OF SENSIBLE TO TOTAL HEAT LINES
DEW POINT TEMPERATURE
DRY BULB TEMPERATURE
WET BULB TEMPERATURE
GRS OF VAPOR PER LB OF DRY AIR
CUBIC FEET PER POUND OF DRY AIR & VAPOR PRESENT
TOTAL HEAT IN BTU PER POUND OF DRY AIR AND VAPOR PRESENT

FIG. 6. RANGE OF PICK UP RATIOS WHICH CAN BE MET WITH MAXIMUM VARIATION OF DEW-POINT WITHOUT REHEAT IF CONDITIONS ARE ALLOWED TO VARY ALONG EFFECTIVE TEMPERATURE LINE †

requires that the dry-bulb temperature vary from about 81.5 F at 40 per cent relative humidity to 78.5 F at 60 per cent relative humidity. The volume of air circulation is assumed to be constant so that variations of dry-bulb temperature and relative humidity result from changes in the proportion of recirculation and dew-point.

† Loc. Cit., see p. 36.

Fig. 6 indicates the ranges of load character which may be handled without reheat when the maintained conditions are allowed to vary along the effective temperature line and the dew-point is made variable from a practical low of 40 F to a maximum of 63.5 F corresponding to the 60 per cent relative humidity point on the effective temperature line. It is shown that by varying the dew-

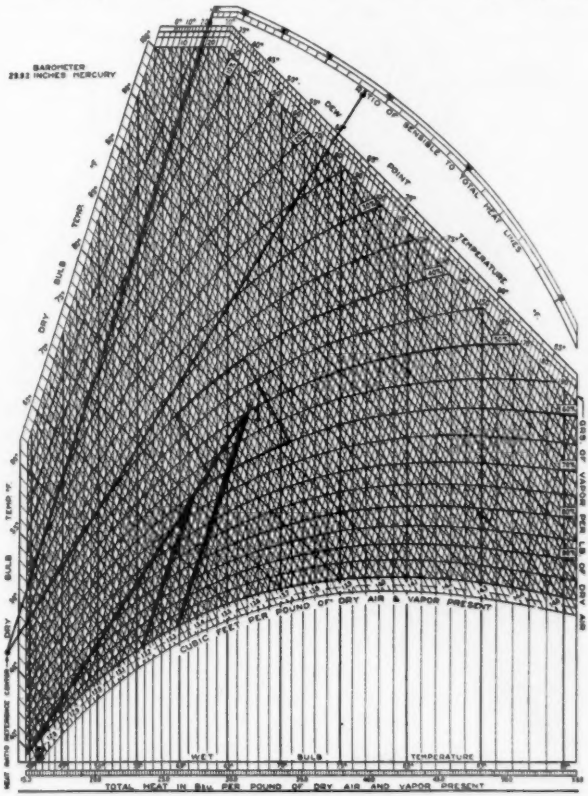
BAROMETER
29.92 INCHES MERCURY
DEW POINT TEMPERATURE
RATIO OF SENSIBLE TO TOTAL HEAT LOAD
DRY BULB TEMP °F
GRS OF VAPOR PER LB OF DRY AIR
CUBIC FEET PER POUND OF DRY AIR & VAPOR PRESENT
WET BULB TEMPERATURE
TOTAL HEAT IN BTU PER POUND OF DRY AIR AND VAPOR PRESENT

FIG. 7. RANGE OF PICK UP RATIOS WHICH CAN BE MET BY VARYING THE DEW-POINT WITH REHEAT IF CONDITIONS ARE FIXED †

point, a change in load character of from 55 per cent sensible heat to 100 per cent sensible heat is possible in maintaining the desired conditions without reheat. By maintaining the maximum dew-point for any given load character, the greatest economy of refrigerating energy will be realized. The range of load character permitted by this scheme of operation is sufficient to meet those changes typical of most commercial applications.

† Loc. Cit., see p. 36.

On Fig. 7 reheat is introduced to the problem. It is assumed that a fixed condition of 80 F dry-bulb and 50 per cent relative humidity is maintained in the conditioned space with variable dew-point and reheat. At the lowest dew-point of 40 F, air by-passing is all that is required and a load character of 63 per cent sensible heat is the only one it is possible to meet. This ratio,

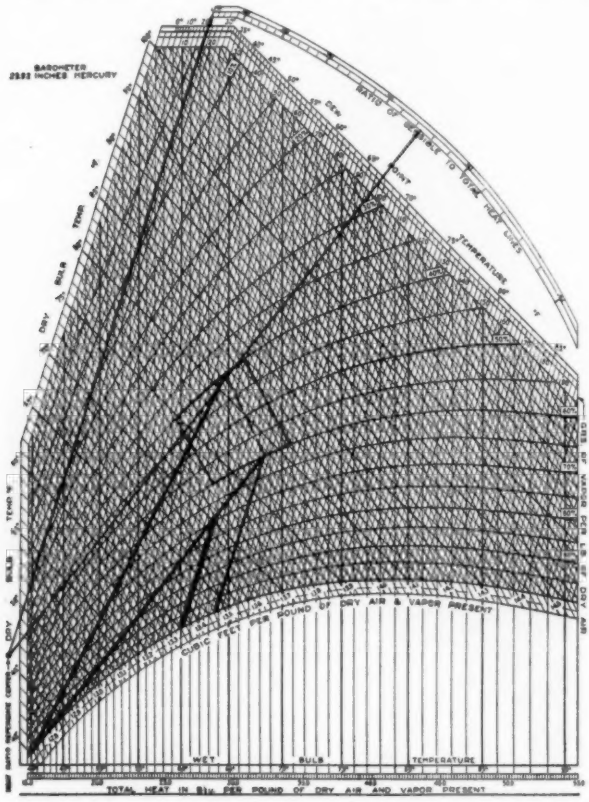
BAROMETER
29.92 INCHES MERCURY
RATIO OF SENSIBLE TO TOTAL HEAT LINES
TEMPERATURE
DRY BULB
WET BULB
CUBIC FEET PER POUND OF DRY AIR & VAPOR PRESENT
GRS OF VAPOR PER LB OF DRY AIR
WET BULB TEMPERATURE
TOTAL HEAT IN B.t.u. PER POUND OF DRY AIR AND VAPOR PRESENT

FIG. 8. RANGE OF PICK UP RATIOS WHICH CAN BE MET BY VARYING THE DEW-POINT WITH REHEAT IF CONDITIONS ARE ALLOWED TO VARY ALONG AN EFFECTIVE TEMPERATURE LINE †

however, can be met by a dew-point of 56 F by reheating from the saturation line to 70 F and eliminating air by-passing. For higher pickup ratios the treatment is the same as for the 63 per cent ratio and the maximum dew-point will rise with each ratio until, at 100 per cent sensible heat pickup, the dew-point is about 60 F. At any pickup ratio between 63 and 100 per cent the use of reheat permits the use of a higher minimum dew-point. Thus, where a

† Loc. Cit., see p. 36.

set of conditions has been assumed which, without the use of sensible reheat, would call for a theoretical dew-point variation of from 40 to 60 F, it is apparent that the same conditions can be maintained by a dew-point variation of only 4 F, from 56 to 60 F, if controlled reheat is provided. It is apparent that large washers are required for these schemes of meeting load character differences but it is believed that the economy of high dew-point operation with steam jet equipment and the ability to maintain desired conditions exactly are factors which may well offset the increase in equipment size.

The range of load character shown in Fig. 7 can be extended by permitting the maintained conditions to follow along the line of effective temperature as shown in Fig. 8. It must be emphasized that sensible reheat from a comparatively high dew-point is required when the sensible heat constitutes less than about 63 per cent of the total heat picked up and the desired conditions cannot be met by continuing to lower the dew-point.

No difference is made in the scheme of operation but the controls would, of course, be different. Some increase in economy of refrigerating energy is possible when the sensible heat ratio is high and the conditions are maintained at the lower end of the effective temperature line.

It has been indicated that the conditions required of all air cooling installations can be met by dew-points within the economical range of a steam jet refrigerator, most of them without reheat and some only if reheat is provided.

It has been indicated also that the attempt to meet the requirements of widely varying heat picked up ratios, such as are met in theaters, ballrooms and restaurants, without reheat and by the use of very low fixed dew-points is not theoretically correct and does not give good practical results. The low fixed dew-point arrangement is apparently an adaptation of mass production equipment to the problem.

It is believed that a simple and correct method of meeting variations in heat ratios picked up is afforded by direct dew-point control which is easily obtained with the steam jet machine.

If the dew-point, and therefore the refrigeration temperature, must be changed to meet exactly those changes in load character which occur in most air conditioning installations, it is important to know the influence of changing refrigeration temperatures on energy consumption. The influence of refrigeration temperature on the economy of steam jet is important so this characteristic is carefully examined in the following paragraphs.

The temperature to which the chilled water is cooled by flash evaporation is termed the flash tank temperature. In some machines, this temperature is held constant by thermostatic control. With such machines, which are designed to cool the circulating water between two fixed temperatures, it is necessary to vary the quantity of chilled water by-passed through the coils or washer to meet variations in load. Other machines may maintain a fairly constant water temperature on the discharge side of the coils or washers by permitting the flash tank temperature to vary, depressing it as required by increases in washer load. With such machines, the quantity of chilled water circulated is held constant, the required variation in refrigerating capacity being obtained by the variation of temperature.

Theory, confirmed by tests of equipment installed on consumer's premises, indicated that the capacity of the steam jet machine to produce refrigeration increased with increase in flash tank temperature. For example, a booster

capacity of 20 tons was obtained when the flash tank temperature was 40 F, and when it was 50 F a capacity of 32 tons was obtained. A little arithmetic will indicate readily why this result was obtained.

The booster system functions to remove water vapor from the flash tank, and to increase the pressure of this vapor to the pressure existing in the condenser where it is condensed, thus permitting the transfer of the heat content of the vapor to available condensing water. The heat of vaporization is withdrawn from the water in the flash tank and this removal results in a lowering of the liquid temperature in the tank. This withdrawal of heat sets up a differential between the heat in the liquid of the water entering the flash tank (H_1) and the water leaving the flash tank (H_2) which differential is made up of the product of the latent heat (L_2) and the fraction evaporated (X).

$$H_1 = H_2 + XL_2 \tag{1}$$

The fraction evaporated (X) is obviously the quotient obtained by dividing the difference in heat contents by the latent heat per pound.

$$X = \frac{H_1 - H_2}{L} \tag{2}$$

The useful heat rejected to or pumped up to the condenser is obviously $H_1 - H_2$.

The tonnage developed per pound of vapor is $\frac{L_2}{12{,}000}$ which corresponds to 11.3 lb of vapor per hour per ton of refrigeration at 50 F.

The specific volume of the vapor formed by the evaporation of part of the flash tank water decreases as the flash tank temperature increases, whereas the latent heat of evaporation remains nearly constant. Thus $\frac{L_2}{12{,}000}$ for 50 F is equivalent to 11.3 lb per ton as given previously or 19,200 cu ft of vapor must be handled per ton; whereas at 40 F, $\frac{L_2}{12{,}000}$ is equivalent to 11.2 lb per ton or to the handling of 27,400 cu ft of vapor per ton. Thus, on the basis of volumes alone, a booster system handling sufficient flash tank vapor to produce 20.0 tons of refrigeration at 40 F would handle enough to produce $\frac{27.4}{19.2} \times 20.0$ or 28.5 tons at 50 F. Actual tests bear out this conception of the jet as a constant volume compressor.

On Fig. 9 are shown the results of tests of refrigeration capacity as a function of flash tank temperature. The capacity of one booster is shown to rise almost as a straight line function of flash tank temperature. The economy of high flash tank temperature operation is shown by the second curve which is almost a straight line function of flash tank temperature. The steam consumption has been expressed as a fraction of the consumption at 40 F flash tank temperature and it is to be noted that these fractions apply exactly only to the machine tested. However, except for variations in the design point, the fractions for other machines will follow the same characteristic curve. It is obvious that the rapid rate of increase of capacity or decrease in steam consumption with increase in flash tank temperature is a characteristic which should be borne in mind in the design of refrigeration for air conditioning.

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The fact that the capacity of a steam jet machine increases with increased flash tank temperature may be taken advantage of either to gain more capacity or to reduce steam consumption. This favorable consumption characteristic is not, of course, peculiar to steam jet compressors but has its parallel in the effect of suction temperature on the capacity of mechanical compressors. In the steam jet system, however, the flash tank introduces a very high thermal storage which makes it practicable to control the refrigerating machine directly, with little chance of hunting, in meeting load changes such as occur in air conditioning work.

In many systems, the chilled water is generated at a temperature as low as 40 F. In order to avoid the complicated systems of regulation which would

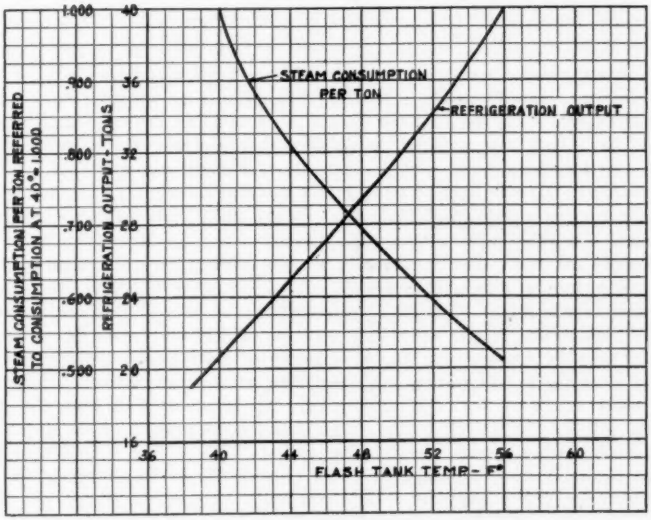

FIG. 9. VARIATION OF STEAM CONSUMPTION AND REFRIGERATION OUTPUT WITH FLASH TANK TEMPERATURE

be required if the refrigeration output were to be instantaneously changed, without lag or over travel, to meet the load changes, a thermostatically controlled valve bleeds the proper amount of chilled water into the dehumidifier circuit to produce the required dew-point. This procedure results in a practical but not particularly economical means of control, which has some understandable advantages from the production and installation standpoint.

The flywheel effect of the flash tank storage makes it possible to control the steam jet unit directly from the dew-point thermostat. This method of control will eliminate the cold water by-pass control and chilled water will be generated at the highest possible temperature compatible with the load.

Physically, the method of controlling the output of the machine is very simple. Usually, the capacity of the machine is divided among several boosters, one or more of which are turned on and off manually or automatically in

accordance with the demand. The normal installation corresponds to a multiple installation of compressors having constant speed characteristics. If the thermostat operating the supply valves to the boosters is the dew-point thermostat, then the steam jet machine will operate only a sufficient time to maintain a flash tank temperature enough below the dew-point to meet the load. As the load increases, the dew-point will tend to rise, but will be restored by the lowering of the flash tank temperature caused by the operation of additional boosters. In this manner, with a fixed dew-point, the flash tank temperature will always be as high as the load permits and the machine will be operating on the point of greatest economy from the standpoint of this characteristic. If the greatest possible economy of operating energy is to be realized, not only must the steam jet machine be controlled by the dew-point directly but the dew-point itself should be maintained as high as possible.

It has been shown that because a steam jet refrigeration machine operates as a constant volume low head pump and because the volume of 1 lb of water vapor decreases rapidly as the saturation temperature or pressure rises without material change in latent heat per pound, it is possible, by raising the saturation or flash tank temperature, to substantially increase the capacity of the machine or to substantially reduce the steam consumption per ton of refrigeration produced. It has been indicated that the thermal storage introduced into the cycle by the supply of chilled water in the flash tank makes it practicable to control the flash tank temperature directly from the dew-point thermostat in a simple manner. The inescapable conclusion is, therefore, that operating energy economy will be greatly improved by operating at the highest possible flash tank temperature. Several specific cases have been discussed to illustrate possible variations of the operating scheme, which will make it possible to take the maximum advantage of the economy of flash tank temperature operation by showing the limitations of fixed dew-point temperature operation and the greater range of load character which can be covered at high dew-points without reheat. With reheat the ranges and economies possible are similar to those obtained without reheat. The authors believe that desired conditions can be obtained most satisfactorily over a wide range by allowing the dew-point to vary under control. The point with respect to steam jet is that whereas the rise of reheat and refrigeration may be set by operating conditions, the rise of saturation or flash tank temperature may be used to cut down the refrigerating energy required by the system over that required by constant flash tank temperature operation. By assuming as a limiting condition that all the air passed through the washer is outside air, it is seen that as the percentage of outside air is increased the offset to increased operating costs by generating chilled water at the highest possible temperature may be considerable.

STEAM SUPPLY PRESSURE

If the pressure of the steam supplied to an operating booster on a steam jet machine is continuously reduced by throttling from the maximum, a point is reached where the jet breaks and refrigeration ceases. A jet will break at pressures which vary with the condenser vacuum, the break pressure decreasing as the condenser vacuum rises. A jet will be stable as long as the steam supply pressure is above the break pressure. Because the steam consumption of a jet is proportional to the initial pressure, it is obvious that to operate

at any pressure materially above the break point is wasteful of steam, and any control that will maintain the supply pressure only slightly above the break point as the condenser vacuum changes will greatly reduce the energy required over that at constant pressure.

To properly develop the technical and practical aspects of this characteristic of a steam jet refrigerating machine, it is advisable to digress slightly to indicate how the steam jet design problem may be analyzed with the aid of a Mollier diagram.

While the following simple analysis of the thermodynamic processes in the main booster is not basically new, it may be advanced to derive an understanding of a basic framework around which jets may be designed. The Mollier diagram, Fig. 10, shows the analysis used. Because certain of the condition points are determined chiefly by experiment, it should be realized

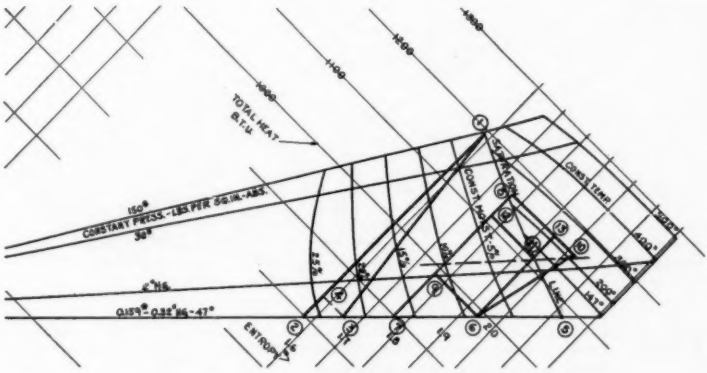

FIG. 10. ANALYSIS OF THERMODYNAMIC PROCESSES IN BOOSTER OF STEAM JET MACHINE

that the analysis is more of a rational method of attack than a complete solution. The line 1-2 represents the isentropic expansion line in the high pressure steam nozzle, while line 1-3 represents the approximate actual expansion line. The energy available for propulsion is measured along the isentropic and is represented by the line 1-4. The two expansion lines, the isentropic (1-4) and the actual (1-3), are used to fix the contour of the main nozzles much as turbine nozzle contours are fixed. The initial condition of the flashed steam or refrigeration vapor may be represented by the point 5 on the saturation line. This flashed steam is mixed with the propelling steam from the jets at the pressure to which the jet steam has expanded. Assuming that the mixing of the two steams is accomplished without losses, the mixture may be represented by point 7 which indicates the increase in unconverted heat content of the jet steam by the admixture of flashed steam. To compress this mixture, available energy is represented by the isentropic compression line 7-8, per pound of mixture. Obviously, some of this energy will be lost in shock losses, etc., before compression starts and that part of the available energy lost may

be represented by that portion of the isentropic between point 7 and 9. On the Mollier diagram, this loss can be represented as a throttling process from point 9 to point 6, giving the initial condition prior to compression as point 6. The final condition may be represented by point 10 which is located by considering that it is on the condensation pressure line and that it has a total heat equal to the heat contents of the propelling steam and flashed steam. Line 6-10 represents the actual path of the mixture compression. Line 11-13 is the available energy lost in compression, friction, shock and leaving loss. Thus, the total available energy of propulsion is accounted for. The condition of the steam at any time during compression is given by the line 6-10. The velocities at points in this path are given by the heat drop along the

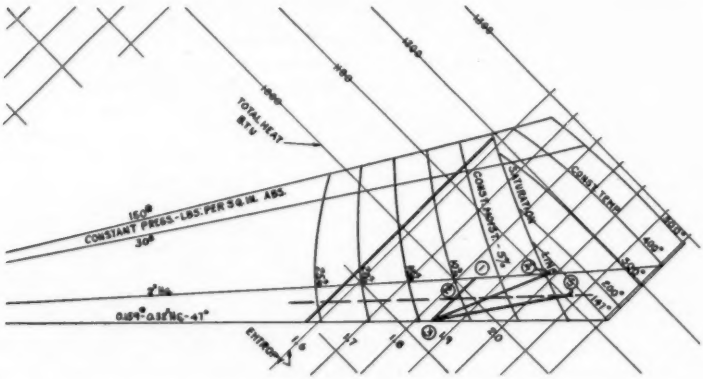

FIG. 11. ANALYSIS OF THEORETICAL WORK OF COMPRESSION AT CONSTANT SUPPLY PRESSURE AND CHANGING CONDENSER PRESSURE

isentropic 13-6 from 13 to the point in question. The compressor cross-section contour may be calculated from these data.

The chart illustrates the energy changes taking place in a booster when the propelling steam acquires a high velocity by expanding to the flash tank pressure, mixes with the flash tank steam and then reconverts some of its velocity energy into static pressure as it pumps the flash tank vapors up to the condensing pressure. It is obvious, after a short examination of this chart, that the energy equations will indicate a lower required steam consumption if the line 6-10 is shortened by raising the flash tank temperature indicated by point 6. Similarly, it is obvious that an improvement in condenser vacuum unaccompanied by other changes will decrease requirements for propelling energy represented by the line 6-10 without decreasing the available amount of such energy. This condition of decreased efficiency is clearly shown on Fig. 11. Considering Fig. 11, it has been previously indicated that the useful work of compression per pound of a mixture at any point along the condenser pressure line is given by an isentropic such as 1-3. Suppose, however, that the condensation takes place at a higher vacuum because the cooling tower is operating under lower outside wet-bulb conditions, which results in conden-

sation along a line of lower pressure. The useful work of compression is now 3-2. The actual compression line is seen to be 3-5 which is much worse than the former lines 3-4. From this, it is seen that the constant pressure operation is done at a considerable sacrifice of real efficiency at partial condenser loads. The explanation rests in the fact that the position of point 3 cannot change as the ratio of propelling steam to flash tank steam is held constant by constant steam supply pressure. The result is that the efficiency, represented by the angle between the actual compression line and the isentropic, must vary with the condenser vacuum and grow worse as the vacuum improves.

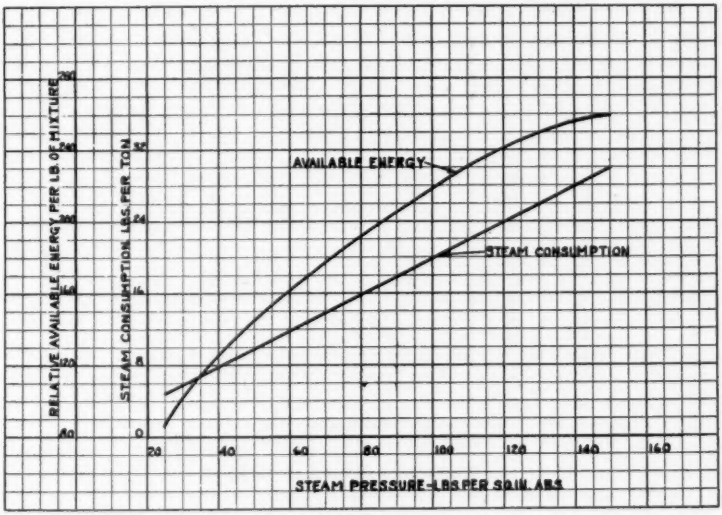

FIG. 12. VARIATION OF AVAILABLE ENERGY FOR PROPULSION AND STEAM CONSUMPTION WITH STEAM PRESSURE

It is possible to correct for this change in condenser vacuum and operate at practically constant thermodynamic efficiency by throttling the steam supply pressure. A throttled steam supply pressure causes the throttle flow and consequently the available energy for propulsion to decrease. Considering the jet as a constant volume compressor, the refrigeration tonnage can be assumed to remain constant while the steam supply pressure is being throttled. Tests of actual machines have shown that the steam supply pressure can be throttled to any point above the break point without appreciable capacity change. Actually, this constancy of capacity is probably due to the fact that the effect of the several changing conditions on the capacity adds up to zero. Thus, the nozzle expansion ratio changes and the nozzle efficiency falls off considerably when the steam supply pressure is varied widely. On the other hand, the shock losses and all velocity functional losses decrease as the initial pressure is lowered, so that in practice, it has been found that the jet acts as

though its over-all thermodynamic efficiency was fairly constant through a considerable range of operating pressure. The effects of a decreased steam flow on each part of the system are beneficial. Thus the cooling tower load decreases until equilibrium approaches and the tower gradually increases its ability to dissipate heat as it comes down on the temperature scale. On the Mollier diagram the effect of throttling the steam supply pressure is to move point 3 along the flash tank pressure line, in such a manner that the actual expansion line for the changed condenser vacuum remains parallel to the design expansion line and no decrease in efficiency takes place.

A measure of the gain made by decreasing the steam consumption by throttling the steam supply pressure is indicated in Figs. 12 and 13. On Fig. 12

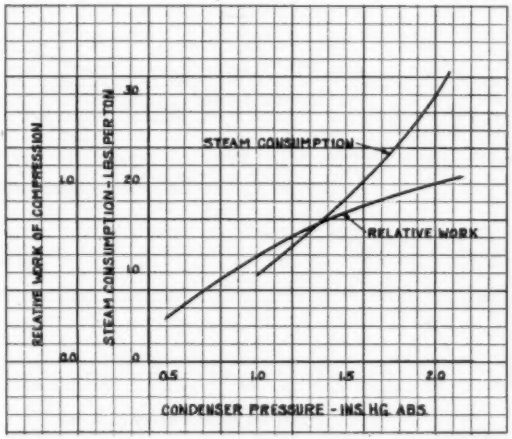

FIG. 13. VARIATION OF RELATIVE WORK OF COMPRESSION AND STEAM CONSUMPTION WITH STEAM PRESSURE

is plotted as a straight line the variation in the steam consumption of a fixed nozzle as the supply is adiabatically throttled from the initial design condition of 150 lb per square inch absolute, initially dry. If for each steam flow and throttled pressure, the theoretical energy made available by throttling and then expanding isentropically to the flash tank is calculated, a curve of available energy for propulsion results which may be drawn on the same chart. This curve indicates, relatively, the energy available for the propulsion of a pound of the mixture of propelling and flash tank steam to any condenser pressure. If more energy is required than is available, the jet will break, but if less is required than is available, that difference is wasted. The energy required, per pound of mixture, is treated as directly proportional to the theoretical work of compression from the flash tank to the condenser pressure at constant efficiency.

On Fig. 13, the relative work done on the flashed steam is plotted against

condenser pressure. The work required to compress from a flash tank temperature of 46 F to a condenser pressure of 2 in. of mercury is taken as unity. By combining the effect of the curve of relative available energy and relative work required, the effect of throttling the steam supply pressure on steam consumption can be indicated. The basic booster steam consumption used was 30 lb per ton of refrigeration, with a flash tank temperature of 46 F and a condenser pressure of 2 in. of mercury. Relatively, the steam consumption will be reduced from 30 to 10 lb per ton when the condenser vacuum rises from 28 to 29 in.

COOLING TOWER AND EVAPORATIVE CONDENSERS

The cooling tower should be considered as the mechanism by which the heat rejected in the refrigeration cycle is finally disposed of to the atmosphere. It may be likened in theory to a large wet-bulb thermometer, the temperature of which is the result of the equilibrium between the gain of heat by the thermometer bulb due to dry-bulb temperature and the loss of heat due to water vaporization from the bulb. The ratio of heat transmission by conduction to that by evaporation is approximately constant over a considerable range. The cooling tower has been analyzed by considering it as a device which operates by virtue of the difference in vapor pressure between the water and the air along the path of cooling. An analogy between temperature difference and vapor pressure difference results from this treatment so that a tower may be assigned a coefficient of heat transmission which dimensionally is Btu per hour per cubic foot per inch of mercury integrated cooling potential.

Since the vapor pressure of water rises rapidly as the temperature of the water rises, the cooling tower is a device which will reject increasing quantities of heat as the water temperature increases, so that the equilibrium between it and the refrigerating machine must always be theoretically attained. Practically the condensing water temperature may not exceed the vacuum conditions under which the jet is stable. Towers are ordinarily designed so that the heat load specified will be rejected at a certain inlet water temperature and outside wet-bulb. The wet-bulb temperature seems to be fixed at 75 F. There are a small number of hours during the year in which the wet-bulb in New York City exceeds 75 F. During these hours when refrigeration is most needed, the machine functioning with a water tower temperature of 75 F will have operating difficulties. There are at least three solutions available which are:

1. The loss of capacity may be accepted at overload as must be done on mechanical compressor installations. This has the psychological disadvantage of being tangibly in evidence.
2. An auxiliary supply of cooling water for handling over peak loads may be cut into the tower circuit. This has the advantage of cutting installation costs at a negligible cost of peak water makeup.
3. The machine itself may be designed for a condensing water temperature which will be adequate to meet any external wet-bulb condition. In this case, it is imperative to include partial load throttling in the design. Since the writers believe that this is the most satisfactory from an all around economy standpoint, this should be elaborated.

In connection with the cooling water problem, the writers have available test data on a 68 ton steam jet unit operating in New York City with an

evaporative condenser. These data are of interest because of the pioneer character of this evaporative condenser and steam jet combination on an air conditioning application. On a day when the outside wet-bulb temperature was 75.4 F the tests show that the machine developed 66.7 tons of refrigeration with a flash tank temperature of 45.7 F. The unit steam consumption was 27 lb per hour per ton of refrigeration.

Since the units are specifically designed and manufactured at present for each installation, there exists the opportunity to engineer more advantages into the installation than may be possible by using off the shelf equipment.

Steam jet coolers need suffer no operating handicaps when compared with other types of equipment. The starting of these machines may be made completely automatic with a single push button operation. They may be controlled for standby operation in connection with adiabatic cooling during the variable months. In this connection, the storage capacity of these machines gives a supply of cold water which may be drawn on for instant use. It is of interest to note that, generally, adiabatic cooling should have a prominent place in any economic study of coil versus air washer installations.

CONTROLS AND REQUIREMENTS

It is quite feasible to consider the completely automatic operation of a steam jet machine and following this procedure through step by step seems a desirable approach to the basic control requirements.

1. Push button starts the cooling water circuit including pumps, towers and fans.
2. Pump circuit is interlocked with motor valve on secondary ejectors.
3. Condensate pump is started by a float switch, the pump output being controlled by a level controller.
4. When the condenser vacuum rises to the proper value, a vacuum switch energizes the main booster control circuit and starts the chilled water pump. The machine should now be on the line.
5. When the chilled water temperature reaches a predetermined setting, a thermostat throws in the throttling control.

The only safety device required practically is one thermostat. It throws the machine off the line should any failure take place which results in loss of load. This consists of a thermostat in the chilled water line set to shut the steam off at an abnormal chilled water temperature. No manual supervision in excess of that required with any other type of apparatus should be needed.

ECONOMY CONTROL

The steam pressure should be throttled in accordance with the condensing pressure so that the tonnage may be delivered at the lowest steam pressure compatible with the jet stability.

One apparatus developed to accomplish this consists of a valve in the steam line controlled by an air operated regulator which automatically balances the pressure differential existing between the condenser and the flash tank against steam pressure, so that every condenser vacuum condition calls for and maintains a proper operating steam pressure. Being a differential device, barometric changes do not affect it.

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A modification of this principle has been successfully employed to control condensing water on purchased water supply. In this application, the steam pressure is held constant and the condensing water is throttled to maintain a constant differential between flash tank and condenser.

It is obvious that two courses are available to the designer of the steam jet in providing maximum economy and ability to handle peak load. It may be shown that certain relationships exist which make it possible to do this without material sacrifice of economy.

1. The stability of geometrically similar jets is proportional to the relationship between the propelling steam and the flashed steam.
2. The stability of the same jet is proportional to the steam pressure.
3. The flow of steam is proportional in the same jet, to the steam pressure.

Two courses are open. In most installations the designer of the machine has considered only constant pressure operation. Consequently, the boosters are designed to give the least steam consumption at the stated condensing water temperature. This has meant designing for the maximum available steam pressure. The result is, and has been, unfortunate. The machines do not perform at peak wet-bulbs without assistance.

On the other hand, if the designer initially gave the machine stability, safely in excess of that to be encountered under peak load, and then provided for throttling of the steam pressure to that required by the actual characteristics of the steam jet, the machine would perform at peak wet-bulbs.

PRACTICAL RESULTS OF THROTTLING STEAM SUPPLY PRESSURE

Studies of the consumption of steam to be expected over the cooling season in New York have shown that constant pressure installations will use, on the average, about 0.6 of the full load steam consumption. The seasonal steam consumption is the product of the following factors:

Installed Capacity (Tons) × Unit Steam Consumption (Pounds per Ton) × Hours of Operation × 0.6 = Seasonal Consumption.

Considering two installations which were both altered from constant pressure operation to variable pressure operation, it was found that the factor 0.6 was reduced to 0.4 or an indicated saving of one third in seasonal steam consumption.

CONCLUSION

The writers believe that the demonstration of the influence of flash tank temperature and steam supply pressure on economy is conclusive and is substantiated by performance records. Where a central station steam supply is available in Manhattan, the operating costs of steam jet are such that it can not be disregarded in any economic set-up. Nor can its simplicity and adaptability to simple control fail to appeal on an engineering basis. Until the present time, concentration has been stressed on the economic production of refrigeration, but in this paper discussion has been extended to the effective and economic use of refrigeration in order that users or potential users of air conditioning may correctly interpret the effect of the dew-point relation they select on the effectiveness and the operating costs of their installation.

RECOMMENDED BIBLIOGRAPHY

General

1. Economic Application of Steam Jet Units, by M. A. Nelson (*Proceedings, N.D.H.A.*, 1936). Gives a background and general exposition of steam jet machines. Well illustrated.

Control

1. Operating Economies Reached by Control of Steam Jet Refrigeration, by A. R. Mumford and Ralph Menke (*Proceedings, N.D.H.A.*, 1935). Steam jet refrigeration discussed from the standpoint of automatic control of water temperature and steam pressure with theoretical and actual performance data. Description of controls.
2. Steam Jet Refrigeration Control by Water Flow Measurement, by J. M. Sperzel (*Proceedings, N.D.H.A.*, 1936). Gives test results and discussion of correlating control on air washers and refrigerating machine. Gives curves of energy, load and wet-bulb duration for New York City.
3. Improving Performance of Steam Jet Coolers, by Charles Hull (*Proceedings, N.D.H.A.*, 1937). Gives carefully compiled consumption records of 100 ton installation using differential condenser water control.

Testing

1. Testing of Steam Jet Coolers under Artificial Load, by A. A. Markson and Ralph Menke (*Proceedings, N.D.H.A.*, 1936). Outlines methods and results obtained by using artificial load technique for field tests. Gives consumption and stability characteristics of 100 ton machine.

DISCUSSION

E. H. TAZE (WRITTEN): The presentation to which you have just listened has given a new perspective on certain phases of air conditioning from an entirely new viewpoint. The authors are not actively engaged in air conditioning work and con-

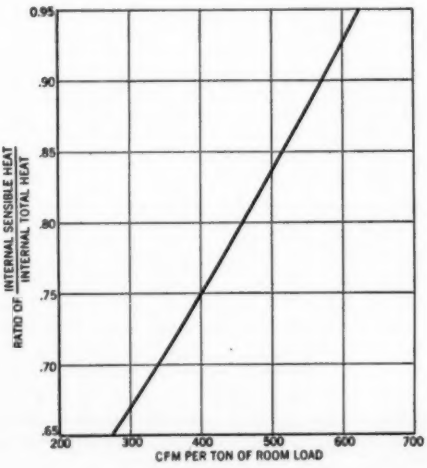
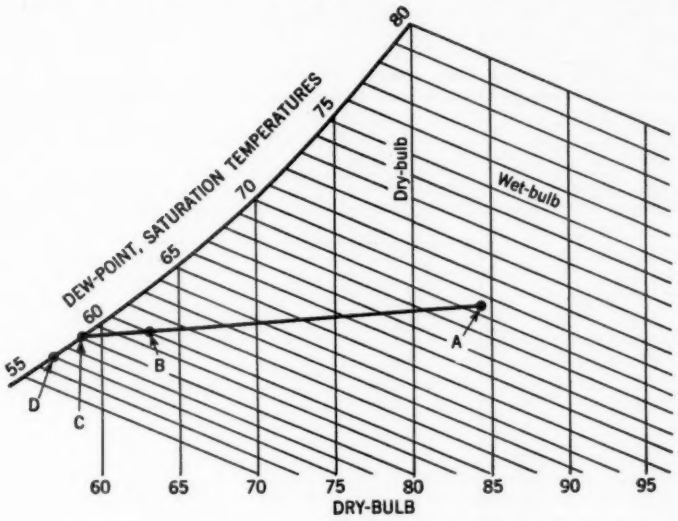


FIG. A. CURVE BASED ON 4 F DIFFERENTIAL BETWEEN DRY-BULB AND DEW-POINT TEMPERATURE OF THE AIR LEAVING COIL. ACCURATE FOR A ROOM CONDITION OF 80 F DRY-BULB AND 67 F WET-BULB ONLY



A = 84 F, D.B.; 69.1 F, W.B.
B = 63.3 F, D.B.; 61 F, W.B.

$C = 59\ F = \frac{\text{W.B. Leaving Water Inlet}}{2}$

D = 57 F = Water Temperature

FIG. B. METHOD OF DETERMINING ENTERING WATER TEMPERATURE FOR COIL WITH AIR TEMPERATURES KNOWN

sequently they have approached this from a purely logical standpoint and brought out some important facts which have not occurred to many of us, I am sure. One of the points which they have brought out is that of the general practice of using unnecessarily low water temperature in a good many air conditioning installations.

Three illustrations have been prepared and it will be noted that Fig. A is a curve showing the relation between the ratio of the internal sensible heat and the internal total heat in any given load condition versus cubic feet per minute per ton of the room load. The values may be calculated by approximate algebraic equation, or with sufficient patience by the cut-and-try method, but this shows a relation between the cubic feet per minute per ton of room load and the ratio of internal sensible to internal total heat, based on 80 deg dry-bulb and 50 per cent relative humidity in the room. It would not hold for any other room condition. I might say further that this is based on coil operation where the dry-bulb and the dew-point leaving the coil are assumed to be 4 deg apart, the dry-bulb of course being 4 deg higher than the dew-point, leaving the coil. From these air volumes and this 4 deg assumed condition, the necessary conditions leaving the coil for any given load condition can be determined very readily.

Fig. B is a section of a standard psychrometric chart, and I have drawn in a line which represents a cooling process based upon air entering the cooler, consisting of 25 per cent fresh air and 75 per cent recirculated air, which would be indicated by this point of approximately 83⅓ dry-bulb and 70 plus wet-bulb. At point *B* I have assumed a point representing, I believe, somewhere in the neighborhood of 90 per cent internal sensible heat to internal total, and from that given air volume, with

the 4 deg spread explained on Curve 1, I have determined this point, which is the necessary condition leaving the cooling coils. That is, the dry-bulb and the wet-bulb cross at this point and that represents the leaving condition. If that line is extended to the saturation curve, a point is reached, which, according to the data of a manufacturer, is the average between the leaving wet-bulb and the entering water temperature required to fulfill the condition. In this particular case the average is 59 deg, which means that the water temperature required is 57 deg. Of course, that obviously is a high sensible load ratio.

A continuation of the process is shown in Fig. C, and I have plotted here points representing required water temperatures entering the cooling coils for various ratios

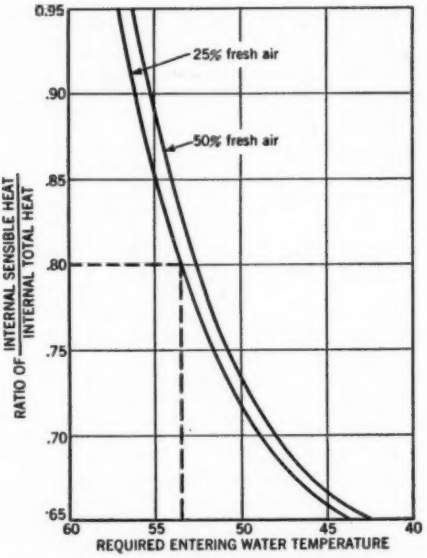


FIG. C. BASED ON ROOM CONDITION OF 80 F DRY-BULB AND 67 F WET-BULB, 31.15 BTU PER POUND. DIFFERENCE BETWEEN TEMPERATURE OF DRY-BULB AND DEW-POINT LEAVING ASSUMED TO BE 4 F

of internal sensible and internal latent load ratios. That process has been continued all the way from 95 per cent down to 65 per cent which reaches the point where reheat is necessary.

It is interesting to note that you can have loads as low as 70 per cent sensible internal load before it is necessary to go below a 50 deg water temperature, and you can go as low as 66 per cent before it is necessary to go below 45 deg water temperature. With 65 per cent sensible heat load in the room it is a high latent load. This bears out one of the points that was brought out in Mr. Mumford's presentation. This same process could be used for air washers, and I will not attempt to go into that at all, but peculiarly enough in air washers a lower water temperature is sometimes required because the manufacturer's standard sizes limit the water volume, so with the dew-point fixed it is necessary to go a little lower on

entering water temperatures than with cooling coils. On these applications it is practical to use the new combination spray coil type of unit for high latent load conditions.

M. A. NELSON[2]: The authors have done a very creditable piece of work and should be commended for the straight-forward method of presenting their thoughts.

In Fig. 9 the authors show a graph illustrating steam consumption per ton of refrigeration and the capacity with change in chilled water temperature. It is assumed that this is the performance of a single booster. There is a word of warning which should be spoken in connection with this figure. It is not always safe to assume that the whole machine will operate in accordance with the data given on this curve, since machines always have been and always will be designed for specific conditions of heat rejection, steam pressure and condenser vacuum. Due to the inherent characteristic of increased capacity with increased temperature, more heat will be rejected to the condenser, and the condenser vacuum is dependent upon the amount of surface and amount of circulating water and its temperature, so that the greater heat load will gradually reduce the vacuum, especially when the unit is operating near the designed capacity. More heat to the condenser water means that the water returning from the cooling tower will be at a higher temperature, further decreasing the condenser vacuum. Finally, an equilibrium point is reached which may or may not be above the safe operating point of the unit.

The amount of overload a unit will safely carry, of course, is also dependent upon variables external to the unit. Perhaps the most important of these is steam pressure. The greater the margin between the minimum pressure available and the designed steam pressure, the greater the overload which can be carried safely. The manufacturer, of course, assumes that the consulting engineer has specified a pressure for design which allows ample margin.

The statements that the steam pressure may be throttled with increasing condenser vacuum without appreciable change in capacity, and that the booster ejecter is a constant volume compressor, are only true over a limited range of flash tank temperatures. Qualitatively, at least, the explanation is that all diffusers are of the converging-diverging type. This is always correct for the design condition and for most operating conditions. However, it is possible to have a condition where the converging portion is no longer required, and under this condition, losses occur which cause changes in the simple relations given in the paper.

There were three questions which occurred to me, and which may have occurred to those of you familiar with steam jet equipment, in connection with the throttling control described. *First,* how does the control operate in case a booster becomes unstable due to foreign material in the nozzle? This is important because when a booster becomes unstable, the unit becomes a heater instead of a refrigerator. *Second,* under conditions of high flash tank temperature, such as exist when the unit is being started, is the control operative or is it necessary to establish normal operating conditions before the control is used? *Third,* does the failure of a single booster in a multiple booster unit cause progressive failure of the remaining boosters?

I should like to repeat that the pressure for which the unit is designed is not under the control of the machine designer. It is a function of the consulting engineer to specify steam pressure low enough to take care of abnormal conditions of wet-bulb temperature when he desires this margin in the refrigeration unit rather than in the cooling tower. When no consulting engineer is employed, it is, of course, the responsibility of the manufacturer to suggest sufficient margin on the steam pressure. It is particularly desirable, from the purchaser's standpoint to obtain all bids on the same steam pressure.

[2] Engr., Steam Condenser Dept., Westinghouse Co., Philadelphia, Pa.

A. R. MUMFORD AND A. A. MARKSON: Mr. Taze's discussion is very much in order and the authors would like to have this discussion considered almost as a part of their paper. Mr. Taze supplies the connection between the apparatus dew-points which for obvious reasons form the basis of the authors' discussion and the chilled water temperatures necessary for his assumed conditions of coil design. Whereas the paper speaks of dew-points, Mr. Taze has taken these and brought them to a basis of actual necessary chilled water temperatures. In every case we see that as the sensible heat decreases in relation to the total heat an impasse is reached where conditions cannot be improved merely by further chilling of the water.

Replying to Mr. Nelson's discussion and his question about the reliability of throttling control in the event of nozzles becoming plugged, the answer is twofold. First, the control may be equipped with a safety device which throws full steam pressure on the machine in case of such emergency. Our actual experience has been that in one or two instances this premonitory condition allowed the operator to correct his trouble before anything more serious took place. Higher steam pressures than necessary, it should be realized, are the crutch upon which a machine may limp along until the hour of reckoning which is likely as not to occur at some time when the weather is most torrid and the machine most needed. Is it not strange then that this positive safety feature of the control should seem opposed to conclusions reached by one or two manufacturers in this field? When it is considered that the use of throttling control has saved from 20 to 35 per cent of the seasonal steam requirements on three large installations in New York City, it may be concluded that such reluctance to advocate throttling of steam pressure is largely unjustified. Further, it would seem in the interests of manufacturers of steam ejectors to recommend the use of a device which enables their product to compare most favorably with condensing turbine drive.

No. 1075

COOLING TOWER EQUIPMENT AND ITS RELATION TO WATER CONSERVATION

By S. I. ROTTMAYER* (*MEMBER*), CHICAGO, ILL.

THE increasing number of cooling-for-comfort installations presents very definite problems to municipalities, especially in metropolitan areas, concerning condenser water supply and disposal. So great has this problem become in certain cities that legislation has been enacted to limit the use of city water for condensers. It follows that water-conserving equipment must be developed for cooling-for-comfort purposes. Such equipment includes means for transferring the heat removed from buildings directly to the outside air using water as the heat-carrying medium.

This paper will deal with a discussion of cooling towers for air conditioning refrigeration, their characteristics and their relation to water conservation, particularly in the Chicago central business district.

Refrigerating apparatus used in cooling-for-comfort installations performs the function of a heat pump in extracting heat from air at a low temperature level and in discharging it at a higher temperature level. Various refrigerants may be employed by the pump for conveying the heat. In the last stage of the transfer, water commonly is used. Where water costs are low, the easiest course has been to discharge this heat-laden water in the sewer. Water cooling towers are devices in which an intimate contact of water with atmospheric air is achieved, thereby promoting an exchange of heat from the water to the air. The water is cooled in the process, to be used again in the refrigeration cycle.

A successful and economical cooling tower must be capable of breaking up the incoming condenser water so as to present the largest possible surface to the air and at the same time bringing into intimate contact with this water the largest possible quantity of air for the least expenditure of power.

The extent of cooling is limited by the wet-bulb temperature of the air. Hence the efficiency of a device as a water cooler is expressed as a percentage as follows:

$$\text{Per Cent Eff.} = \frac{(\text{hot water temperature} - \text{cooled water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of incoming air}}$$

Water cooling towers may be classified as:

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1. The atmospheric type which depends on wind motion for operation.
2. The natural draft type, which employs stack or thermal action for producing air movement through the tower.
3. The mechanical draft type, which employs forced or induced draft fans for creating air movement through the tower. The humidifying air washer is a modification of an induced draft cooling tower.
4. The evaporative condenser, which is a combination of a refrigerating condenser and a mechanical draft cooling tower.

EQUIPMENT CHARACTERISTICS

The *atmospheric tower*, called also open natural draft, deck type or wind tower and illustrated in Figs. 1 and 2, is the most common type of tower

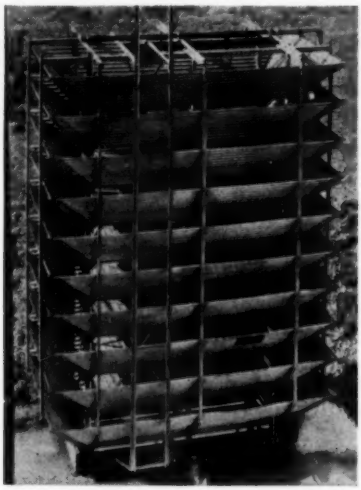

FIG. 1. AN ATMOSPHERIC COOLING TOWER FRAME; LOUVERS AND COLLECTING BASIN ARE MADE OF STEEL

installed for general refrigerating purposes, considering the number and tonnage of installations. It is constructed of wood or steel framing, adequately braced for wind stress, supporting a filling consisting of a series of horizontal decks or splash bars equally spaced from top to bottom. A system of small louvers or large inclined shields at each deck level is supported by the outside of the framing. The water enters at the top through troughs, splash heads or spray nozzles and descends from deck to deck to reach a basin at the foot of the tower. The function of the louvers or inclined shields is to prevent loss of water on the leeward side by deflecting the air upward, retaining the water particles blown against them. Vertical, staggered eliminators between the side shields also have been used to reduce further the windage losses or drift of the water.

Increasing the height and length of a tower increases the area of water

exposed to air, and increased width increases the length of time of contact of the air with the water.

Since this type of tower depends for operation on natural air movement, it must not be located near obstructions which reduce wind contact and its

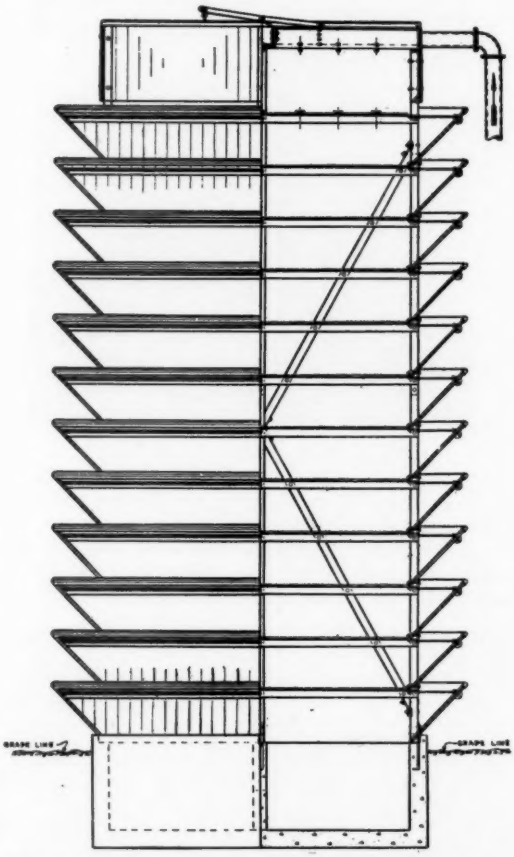

FIG. 2. HALF-SECTION AND ELEVATION, SECTIONAL PORTABLE ATMOSPHERIC COOLING TOWER

long side must be at right angles to the prevailing winds. It should be designed to operate at capacity with a wind velocity of not more than 5 mph and with a wet-bulb temperature which is not exceeded during more than 5 to 8 per cent of the season.

Atmospheric spray towers shown in Figs. 3 and 4 are a modification of this

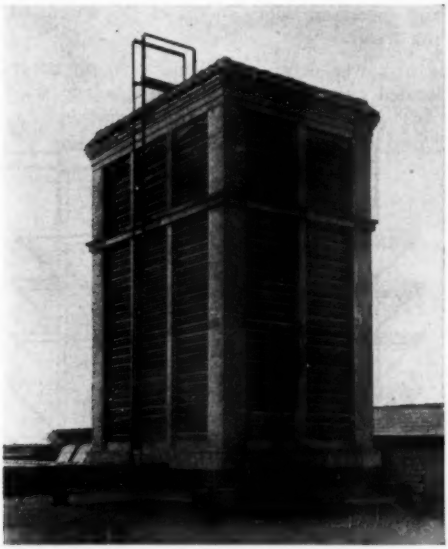

FIG. 3. ATMOSPHERIC SPRAY TOWER

type of tower in that the filling or decks are omitted and water is sprayed downward under pressure from nozzles.

Atmospheric towers have advantages of low operating costs and low first cost. Disadvantages are the large size, great area and weight, as well as dependence on variable natural air movement. Spray towers require a higher

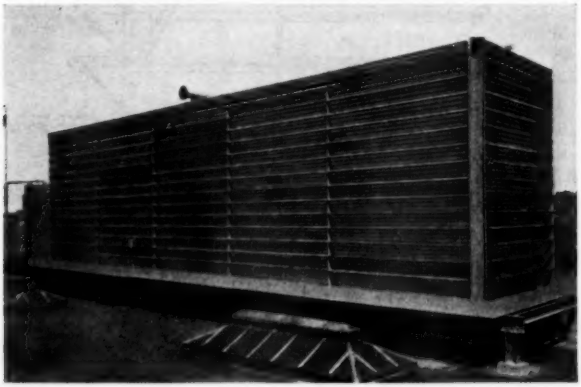

FIG. 4. AN ATMOSPHERIC SPRAY TOWER OF ALL STEEL CONSTRUCTION

pumping head and generally are lower in first cost than purely atmospheric towers.

The efficiency of an atmospheric cooling tower will vary with its design and loading usually between limits of 35 and 90 per cent. The usual atmospheric deck type tower has an efficiency of about 67 per cent compared with about 50 per cent for an atmospheric spray tower.

The *natural draft tower* has an air-tight shell enclosing the filler section with air inlets at its base and with its top extending as a chimney above the

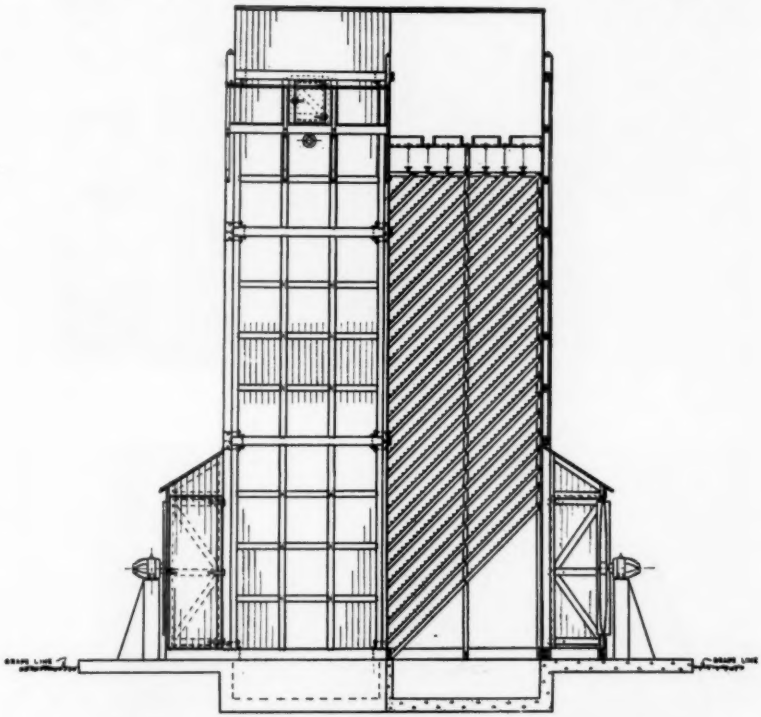

FIG. 5. HALF-SECTION AND ELEVATION, FORCED DRAFT COOLING TOWER

filler for a considerable distance. An upward flow of air from the bottom through the filler is produced by the difference in weight of air on the inside and outside of the tower, this weight responding to temperature; hence its use is confined largely to application having high entering water temperatures and long cooling ranges. Its performance is not greatly affected by wind velocity.

A *mechanical draft cooling tower* differs from the natural draft type in that the chimney above the enclosing shell is shorter and air is circulated upward by means of fans located either at the bottom or at the top. The

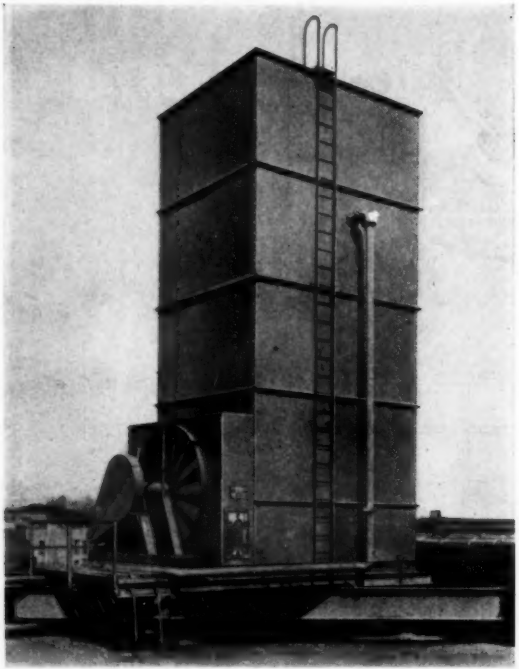

FIG. 6. USUAL STYLE OF AN ALL STEEL FORCED DRAFT COOLING TOWER

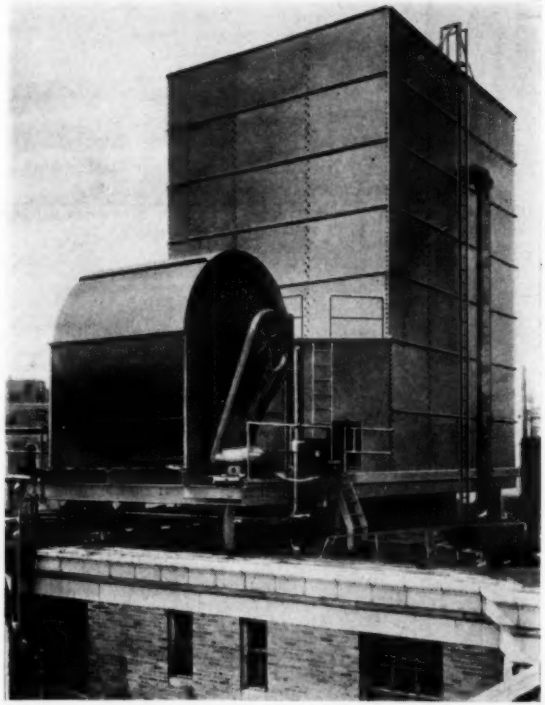

FIG. 7. A FORCED DRAFT COOLING TOWER INSTALLED ABOVE HOTEL GUEST ROOMS

water may fall downward trickling over a wood checker work filler or may be sprayed downward inside the shell without any filler. A set of eliminator plates across the top of the tower is provided for removing entrained moisture from the departing air.

A forced draft tower has fans located at the base of the tower as illustrated in Figs 5, 6 and 7, where they can readily be inspected and serviced. Horizontal vanes may be installed inside the tower to deflect the incoming air evenly across the tower area. An induced draft tower, as indicated in Fig. 8, has fans located above the eliminator plates at the high point of the tower. The fans in such a tower are subjected to the deteriorating effect of the moist

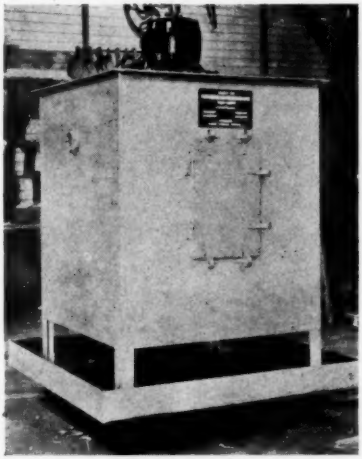

FIG. 8. AN INDOOR INDUCED DRAFT COOLING TOWER

air. This tower can be lower in height than the forced draft style for the same cooling results.

Since the air flow is counter to the water flow, the efficiency of heat transfer of the mechanical draft tower is high. Air velocities over the gross area of the mechanical draft tower generally vary with the design of the tower between limits of 250 and 600 fpm.

Mechanical cooling towers may be located within buildings or may be exposed outside. In many instances they are placed outside as represented in Fig. 9, and are concealed by pleasing architectural treatment. When installed within buildings, mechanical cooling towers may serve as exhaust fans for the ventilating system and may take all or part of the air from inside the building. This air may be cooler and more favorable for service than the outside air.

An ordinary double spray humidifying air washer may be used within a building as a water cooler where considerable area but little head room is

available. The high efficiency of counter-flow heat transfer by the vertical mechanical draft tower is not present in this type of apparatus; hence a greater quantity of air must be handled for the same duty.

Disk or propeller type fans generally are used for producing air circulation in mechanical water towers located outside of buildings. Increasing the number of blades of these fans decreases the noise produced. Where particularly

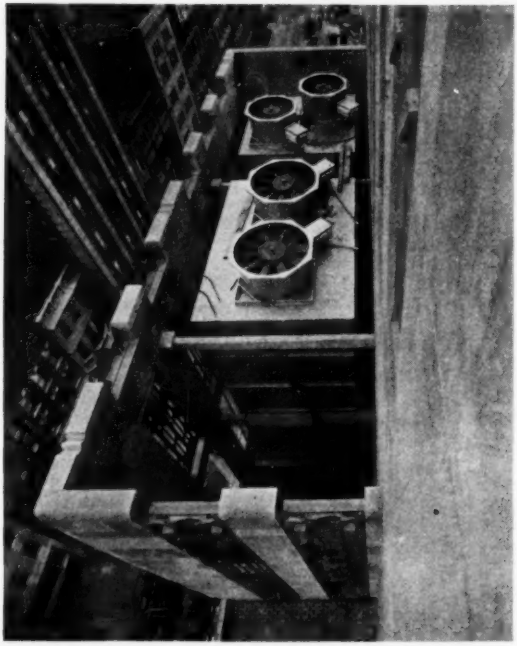

FIG. 9. INDUCED DRAFT MECHANICAL COOLING TOWERS CONCEALED WITH ARCHITECTURAL TREATMENT

quiet operation is essential, housed centrifugal multiblade blowers usually are employed.

Advantages of mechanical draft towers over atmospheric towers are independence of natural wind movement, smaller area and weight. Disadvantages are higher investment, maintenance and operating costs.

The efficiency of a mechanical draft cooling tower ranges between 35 and 90 per cent, depending on its design and loading. The usual design has an efficiency of about 67 per cent.

The *evaporative condenser*, as diagrammatically shown in Fig. 10, combines an induced or forced draft cooling tower with a refrigerating condenser. Like an indoor cooling tower, it may take all or part of its air supply from inside the building, and hence may serve also as an exhaust ventilating fan. It is

constructed generally with a single bank of spray nozzles wetting the extended coil surface of a refrigerant condenser built into the air passage. A bank of eliminator plates is interposed between the spray chamber and the leaving air stream.

The counterflow principle of heat transfer may be incorporated in this design. For periods of intermediate temperature during the spring and fall this apparatus may be operated without water as an air-cooled condenser.

An evaporative condenser, as its name implies, secures a higher heat transfer rate from the refrigerant to the air than is possible with any arrangement using water alone. This is due to exposure to the cooling air not only of the water but also of the tubes containing the liquid refrigerant. This liquid is many degrees warmer than the air and also is several degrees warmer than the water, and due to a greater temperature difference between the donor of the heat and its recipient, the combined heat transfer rate increases measurably. Much more water is also evaporated per unit of refrigeration, and in addition advantage is taken of the latent heat.

Evaporative condensers usually have been made in comparatively small factory-assembled types, but no particular size-limit needs to exist. They are used with ammonia and the fluorine group of refrigerants having a condensing temperature not warmer than about 100 F.

MAKE-UP WATER REQUIREMENTS

The quantity of make-up water required by each type of air-contacting equipment is equal to that lost by wind or drift plus that evaporated. The loss in pounds of water through evaporation is equivalent to the quantity of heat to be dissipated, divided by the latent heat of vaporization at the temperature of evaporation, usually slightly above the wet-bulb temperature.

The heat removed per minute in a refrigeration system is 200 Btu per minute per ton plus additions due to leakage, friction, etc., and plus variable additions on account of the ratio of compression, character of the refrigerant, etc.

In the illustrative examples, conditions such as exist in the locality of Chicago will be assumed.

1. Design wet-bulb temperature—75 F.
2. Heat dissipation per ton mechanical compression dichlorodifluoromethane refrigerating system—250 Btu per minute.
3. Heat dissipation per ton, steam jet refrigerating system—737 Btu per minute.
4. Maximum design city water temperature—76 F.

The water loss due to evaporation, being a function of the heat dissipation, may be expressed as:

$$\text{Gallons per ton evaporative loss} = \frac{\text{Btu per ton heat dissipation}}{8.33 \times \text{latent heat of vaporization at temperature of evaporation}}$$

The temperature of evaporation may be taken at 3 F above the wet-bulb temperature.

For the compression system of refrigeration, the evaporative loss will be:

$$\frac{250}{8.33 \times 1094.3} = 0.0275 \text{ gpm per ton}$$

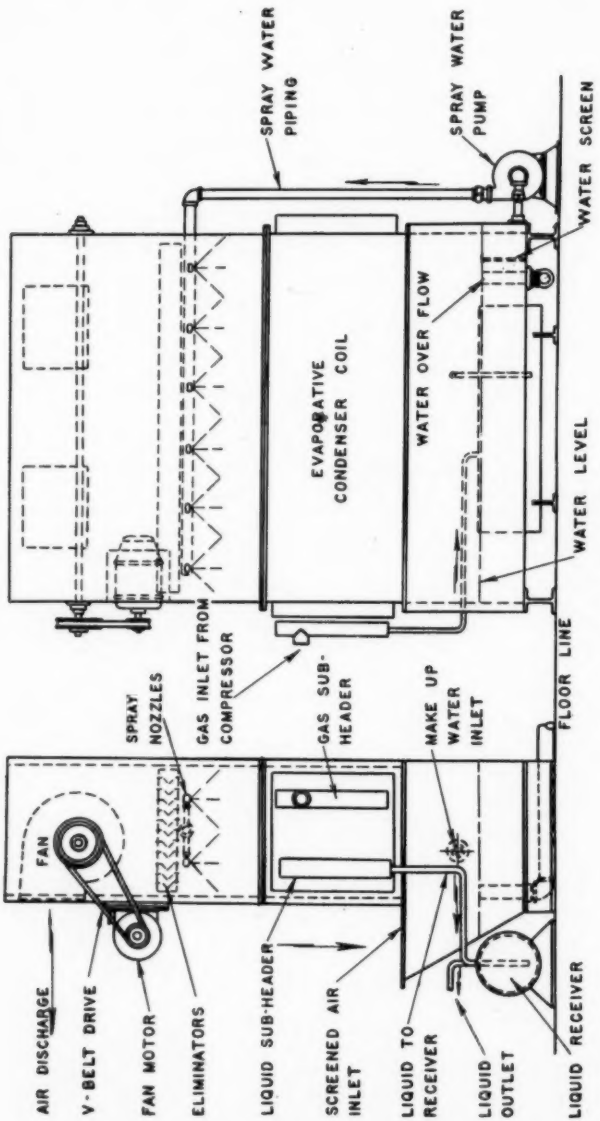

FIG. 10. AN EVAPORATIVE CONDENSER

and for the steam jet system:

$$\frac{737}{8.33 \times 1094.3} = 0.08 \text{ gpm per ton}$$

For a dichlorodifluoromethane mechanical compression refrigerating system, with a wet-bulb temperature of 75 F, condenser water temperature range, 85 F to 95 F, the water circulated per ton will be 3 gpm. A cooling range of 7.5 to 10 F is usual with 75 F wet-bulb. The drift loss then is 1 per cent of 3 gpm or 0.03 gpm.

On the basis of 80 F design wet-bulb and a cooling tower of 67 per cent efficiency, then the water circulation required for the steam jet system is:

$$\frac{737}{8.33 \times (95 - 85)} = 8.85 \text{ gpm per ton}$$

The drift loss then will be 0.0885 gpm per ton.

The total evaporation and drift losses per ton of refrigeration, for the conditions stated, are given in Table 1.

Ordinarily it is considered that in atmospheric cooling about 1 gal of water per 1000 gal of water circulated per degree of cooling range is lost by evaporation and 1 gal of water per 100 gal of water circulated is lost by drift. Thus for a plant in which the water enters the cooling tower at 88 F and departs at 80 F and which circulates 500 gpm, the loss will be $\frac{8 \times 500}{1000} = 4$ gpm by evaporation and $\frac{500}{100} = 5$ gpm by drift, or a total of 9 gpm.

If city water is to be used for condensing purposes, with the warmed water wasting to the sewer, the water consumption per ton of refrigeration will be:

Mechanical compression refrigerating system—

$$\frac{250}{8.3 \times (95 - 76)} = 1.58 \text{ gpm per ton}$$

Steam jet refrigerating system—

$$\frac{737}{8.3 \times (95 - 76)} = 4.65 \text{ gpm per ton}$$

The saving in water consumption if a cooling tower should be used then will be:

$$\frac{100(1.58 - 0.0575)}{1.58} = 96.5 \text{ per cent for the compression system}$$

and

$$\frac{100(4.65 - 0.1685)}{4.65} = 96.9 \text{ per cent for the steam jet system}$$

A value of 90 to 95 per cent generally is taken as the water conservation effected by the use of cooling tower equipment.

TABLE 1—EVAPORATION AND DRIFT LOSS

REFRIGERATING SYSTEM	EVAPORATIVE LOSS, GPM PER TON	DRIFT LOSS GPM PER TON	TOTAL LOSS GPM PER TON
Dichlorodifluoromethane Compression	0.0275	0.0300	0.0575
Steam Jet	0.0800	0.0885	0.1685

Steam jet refrigeration is especially favorable when cooling tower water is available. As the load varies, following changes in the outdoor condition, the compression ratio between the condenser and the evaporator can be reduced, directly reducing the steam consumption and the demand upon the cooling tower. This wide flexibility of output without corresponding loss in efficiency militates in favor of the steam jet-cooling tower combination, even though a steam jet system requires about three times as much cooling tower condensing water capacity per ton as does a dichlorodifluoromethane compressor system. The latter type when under as little as 25 per cent load, a situation which occurs during many hours, usually requires at least 40 per cent of the full power input. With the steam jet type the efficiency really increases as the load is reduced when cooling towers are employed.

Where water cooling tower equipment is used with steam jet refrigeration it is usual to base the design of the tower on a wet-bulb temperature of 3 to 5 F higher than ordinarily is used. This is because steam jets will *flash-back* and fail entirely to function properly if the leaving condenser water exceeds the temperature for which it was designed. The allowance of 3 to 5 F serves as a safety factor for operating insurance.

Carbon dioxide compression refrigeration is not practicable with cooling towers except possibly in exceedingly dry climates, as the critical temperature of this refrigerant is 88 F and the power cost becomes excessive unless the condensing water enters cooler than 80 F. Dichlorodifluoromethane, ammonia and water vapor can be condensed with reasonable success with water which leaves the cooling tower at 85 F.

In the central business district of Chicago, bounded on the North and West by the Chicago River, on the South by Twelfth Street, and on the East by Lake Michigan, there were 16,328 tons of refrigeration installed for air conditioning purposes on October 1, 1937. Of this, only six installations, comprising 620 tons, are provided with water cooling tower equipment; the remaining 15,708 tons use city water for condensing and waste the warmed water

TABLE 2—TABULATION OF COMPARATIVE COSTS PER TON OF REFRIGERATION

TYPE OF WATER COOLING	FIRST COST IN DOLLARS	FLOOR SPACE IN SQUARE FEET	WEIGHT IN POUNDS	OWNING AND OPERATING COSTS IN CENTS PER HOUR
City Water	2	0	0	0.660
Forced Draft Cooling Tower	40	1	160	0.719
Evaporative Condenser	48	0.65	75	0.782

to the sewer. Of the six installations, one, a 70 ton plant, was installed in 1935 to use city water, a second, a 200 ton plant, was installed in 1936 to use city water. In 1937 forced draft cooling towers were installed for both these plants after it was found necessary during certain periods of the previous year's operation to stop the equipment because of inability of the sewers to carry the condenser water. The third and fourth plants of 40 and 10 tons capacity, respectively, were installed with evaporative condensers in 1937. The fifth, a 200 ton plant, was installed in 1937 with a forced draft cooling tower, and the sixth is a low pressure steam jet plant which has been in service for three years with a forced draft cooling tower.

The facilities for supplying water to this central business district are adequate to serve all estimated demands for the next 10 years. The cost of water, 51 cents per 1000 cu ft, is lowest of any city in which there is available information. The sewer facilities in much of the district, however, are hopelessly inadequate and there are no prospects for their improvement. This situation, therefore, of itself requires that any air conditioning refrigerating plant, if it shall be capable of operating whenever required, shall have its own water cooling tower equipment for dissipating the extracted heat to the atmosphere rather than to the sewer.

The data in Table 2 represent approximate relations per ton of refrigeration for comparative purposes. Exact costs, of course, will depend on the size and specific conditions of the job. Assumptions made for these illustrations, in addition to those in the foregoing examples, were: dichlorodifluoromethane compression refrigerating system, all metal construction of cooling towers, 2.5 cents per kilowatthour electricity cost; 51 cents per 1000 cu ft city water cost; 1200 hours operation per season, interest and depreciation computed at 10 per cent for owning cost.

Independent analyses have shown that with present equipment costs and water costs for a load under 600 tons, the cost for the city water-using plant is slightly less than that using cooling tower equipment, and for loads above 600 tons there is a slight advantage to owning and operating cooling tower equipment. An increase in water rates, which might occur should a city filtration plant be installed, and a decrease in cost of cooling tower equipment after its wider usage and development, might easily change the economic balance.

DISCUSSION

J. R. HERTZLER (WRITTEN): Mr. Rottmayer is to be complimented upon an intelligent analysis of a situation which has been given undue publicity in recent months based largely upon inaccurate analyses made by public officials as to the water demands of air conditioning systems as installed for comfort cooling.

In the paper reference is made, however, to certain items of equipment and comparative performance is indicated which we believe could be further qualified for the sake of accuracy.

Reference is made to the use of double spray humidifying air washers for water coolers within a building and the author states that "the high efficiency of counterflow heat transfer by the vertical mechanical draft tower is not present in this type of apparatus; hence a greater quantity of air must be handled for the same duty". It should be pointed out here that were the double spray humidifying air washer arranged in stages to approach the space requirements of a vertical forced draft

cooling tower, comparable performance could be obtained. Commercial apparatus is now in the market which makes comparable performance possible handling no greater quantities of air than would be required for the customary vertical mechanical draft tower. References are made in the paper to comparative operating cost between steam jet refrigerating systems and systems of the positive compression type using dichlorodifluoromethane as the refrigerant.

We would refer the author to THE A.S.H.V.E. GUIDE 1937, Problems 5 to 9 at the end of Chapter 11, pp. 214 to 216 inclusive. In these problems in THE GUIDE is established the method of comparing the overall operating cost of a steam ejector system with a dichlorodifluoromethane system. This problem analyzes the complete operating cost including the operating cost of the cooling tower with all of its auxiliaries and since the power requirements of the cooling tower will vary with a steam jet system as compared to a dichlorodifluoromethane system, this should rightfully be taken into consideration. We believe that the writer of the paper is in error when he makes the statement that a dichlorodifluoromethane system "when under as little as 25 per cent load, a situation which occurs during many hours, usually requires at least 40 per cent of the full power input. With the steam jet type the efficiency really increases as the load is reduced when cooling towers are employed."

Question 7, heretofore referred to as included in THE GUIDE 1937, p. 215, shows an electrical input to the refrigerating system using dichlorodifluoromethane as the refrigerant of 1.1 kw per ton at 284 tons and a kilowatt input of 0.59 kw per ton at 79 tons, which is approximately 25 per cent capacity.

If a given refrigerating system is to be called upon to operate at 25 per cent capacity for any large period of time, the system should be designed so that economical operation can be obtained at that percentage of maximum capacity. It is customary in large installations utilizing dichlorodifluoromethane as the refrigerant to install more than one compressor in a given system making the full amount of evaporator surface and condensing surface available for all conditions of operation.

Information is included in a previously published paper[1] which shows the effect of properly designing a refrigerating system to take advantage of the full amount of evaporator and condenser surface installed to meet maximum heat gain demands. If for instance with the existing cost of power and water available for a given installation, a given amount of condenser surface were justified to obtain under conditions of maximum load a condensing temperature of 110 deg, Fig. 5, p. 418 of the paper referred to would indicate that if the same amount of condensing surface were operated at ½ capacity, the condensing temperature would be reduced 5.5 F.

If a given evaporator were designed originally for a temperature difference of 10 F between, let us say, the water being cooled which is later used for dehumidification purposes and the evaporator temperature; at ½ of maximum capacity, the mean temperature difference could be 5 F and therefore the evaporator which at maximum load would require an evaporator temperature of 40 deg can at ½ load operate at a 45 F evaporator temperature. Thus, whereas at maximum load the operating conditions of 40 deg evaporator and 110 deg condensing temperature were economically justified, the change to 45 deg evaporator temperature and 104.5 deg condensing temperature would involve in accordance with Fig. 4 of p. 417 of the paper referred to a reduction from 98¢ per ton per hour operating cost based on power at 1¢ per kilowatthour to 8¢ per ton per hour which indicates reduction of 18.3 per cent made possible by the effect of doubling the evaporator and condenser surface for partial load operation. Thus, even without changing the temperature of water leaving the condenser which would be true on a city water installation operated from a thermostatically controlled water regulating valve, a power reduction of 18.3 per cent could be obtained with a dichlorodifluoromethane system provided the compressor flexibility were furnished to operate at the reduced capacity. Where,

[1] Application Factors Which Govern the Selection of Refrigerating Equipment for Air Conditioning Service, by J. R. Hertzler (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 411).

on a cooling tower system, reduced load operation is likewise accompanied by a reduced wet-bulb temperature of the outdoor air which will make it possible to supply colder water from the cooling tower, comparable reductions in kilowatt input per ton to the refrigerating system to that given in THE GUIDE are obtainable. The figures quoted were 1.1 kilowatt per ton at maximum load and 0.59 kilowatt per ton at 25 per cent load.

Briefly, the operating characteristics of a dichlorodifluoromethane system when utilized with a cooling tower are such that definite reductions in power requirements of the compressor motor are obtained when colder water is made available from the cooling tower system. The steam ejector system has no advantages over the properly designed dichlorodifluoromethane system in this respect.

J. DEB. SHEPARD (WRITTEN): Mr. Rottmayer's description of equipment and technical discussion is complete and no further additions seem necessary. I would, however, like to take this opportunity to present some statistical information relative to the application of water conservation equipment in the Baltimore area. During the past two years, at least 22 air conditioning installations, representing a total capacity of 1280 tons, have been equipped with evaporative condensers. In addition to these, four air conditioning installations using compression refrigeration, with a capacity of 600 tons and one 300-ton steam jet system, have been equipped with cooling towers. Approximately 20 per cent of the total tonnage used for air conditioning purposes in the Baltimore area uses water conservation equipment. With respect to evaporative condensers, individual single installations run in capacity from 10 to 250 tons, which confirms Mr. Rottmayer's statement that there need be no particular size limit to evaporative condenser applications.

I believe it will be interesting to note some of the factors which have been responsible for the use of condenser water conserving equipment. The Municipal Water Bureau has publicized the serious problems of supplying large quantities of condenser water for air conditioning purposes. If the present safe water supply capacity of the City of Baltimore is exceeded, providing additional capacity will necessitate the creation of a costly new reservoir up the water shed beyond the existing reservoirs. They point out that the summer revenue derived from the sale of water for condensing purposes, at the present rate, can never amortize the cost of an additional reservoir. The situation is particularly serious because air conditioning water use imposes a load on the normal peak consumption and during the period of lowest rainfall. As yet there has been no move to increase water rates or restrict the use of water for air conditioning purposes.

The problem is not entirely one of available supply but also one of distribution and disposal of water. The Water Bureau has shown no disposition to increase its local distribution equipment simply to handle air conditioning requirements, so that in certain sections where loads are high, users are beginning to experience difficulty with supply pressures and are thinking of installing equipment which will make them independent of the City supply. Furthermore, municipal ordinances make it necessary that all uncontaminated discharge water, such as comes from condensers, be disposed of through the storm water sewage system, some parts of which are overloaded during heavy rains so that shut down of equipment is occasionally necessary.

The average large water consumer in Baltimore can purchase water for condensing requirements at 75¢ per 1000 cu ft and the normal temperature of supply is such that the net water cost will be about equal to that given by Mr. Rottmayer for Chicago. Where ample water supply and drain facilities are available adjacent to refrigeration equipment, it is difficult to economically justify the installation of water conservation equipment. However, where it is necessary for the prospective user of air conditioning to provide at his own cost new water service or extension of storm drain facilities, the cost of this work will generally go a long way toward covering investment in water conservation equipment. Even where this is not the

case, many purchasers of air conditioning equipment feel that the situation is such that water rate increases are bound to come and will make the investment a paying proposition. Others feel that should an extremely dry period reduce the normal water supply, restriction will be quickly placed on air conditioning use. Therefore, they feel that the installation of such equipment is necessary to protect their investment in air conditioning and assure its continued operation regardless of restrictions.

These various factors, together with aggressive sales efforts on the part of air conditioning contractors, have resulted in a wide use of water conservation equipment. As the situation now stands, it seems safe to say that most future installations of 40 tons or over and many of smaller size will be equipped with water conservation systems.

PROF. C. F. KAYAN[2] (WRITTEN): As a resumé of the different methods of cooling condensing water this paper serves well to give a picture of the cooling problem and its economic phases.

To members of the A.S.H.V.E. it would probably be still more interesting to have additional discussion of the indoor cooling tower, the concealment by architectural treatment of the outdoor cooling tower, and the evaporative condenser. On the latter it should be pointed out that whereas the straight liquid cooled condenser with forced circulation has a fairly high rate of heat transfer in Btu per square foot per degrees Fahrenheit per hour, and a dry air-cooled condenser a relatively low rate of heat transfer, the evaporative job stands between and simultaneously performs the role of circulating water cooler and condenser. The sheet of water on the external tube surface abstracts heat from the higher temperature refrigerant within the tube and tends to warm up; simultaneously in its downward passage over the tube bank it tends to cool by evaporation due to a temperature and vapor pressure higher than that of the air wet-bulb. Its temperature is therefore intermediate between the condensing temperature and the wet-bulb, and this with its attendant flow rate over the heat transfer surface contributes measurably to an enhanced heat transfer.

On p. 65 the author makes the statement that an induced draft cooling tower can be made lower in height than the forced draft style for the same cooling results (possibly because of fan location). It would be interesting to know why this is the case.

Further elucidation of the comparative calculations for evaporation and drift loss on pp. 67-69 could serve to clear up obscurities. The title for Table 2 could likewise be amplified, particularly with reference to First Cost in Dollars. It is not entirely clear as to what is meant, though one could presume that Condensing Equipment Cost is intended.

Finally, the value of this interesting paper could be still further enhanced if additional specific information were supplied on the cooling efficiencies as a function of its major variables, since limits of 35 to 90 per cent are rather extensive in range, the author citing a customary design figure of 67 per cent as a middle value.

A. R. MUMFORD:[3] There are two or three points in the paper to which the indicated answers may vary in accordance with changes in the imposed conditions.

Table 1 shows evaporation and drift losses which indicate that steam jet has water losses approximately three times those of compression systems. If the condensate from the condenser of a steam jet system is introduced into the tower circuit, no evaporation and drift loss make-up is required. The condensate is more than sufficient to offset such losses.

On p. 69 the comparison of the city water required for condensing is, I believe, based on the same condenser temperature range for steam jet as for mechanical compression. The full condensing range of steam jet is not taken advantage of unless the 95 deg upper limit is fixed by the Sewer Department. In New York

[2] Prof. Mech. Engrg., Columbia University, New York, N. Y.
[3] New York Steam Corp., New York, N. Y.

we are allowed to discharge water to the sewer at 100 deg. That 5 deg extra range, which is possible with steam jet and may be possible with compressor, although the compressor may be restricted by other factors, would permit approximately a 20 per cent reduction in the figure given for the gallons required per ton of refrigeration.

Similarly, on p. 70 in the paragraph which states, "Where water cooling tower equipment is used with steam jet refrigeration, it is usual to base the design of the tower on a wet-bulb temperature of 3 to 5 F higher than ordinarily is used" we point to remarks of Mr. Nelson, who asked consulting engineers to consider the manufacturers when asking performance figures. If the engineers ask for performance figures based on peak conditions, they are going to get them and they will be the best figures the manufacturer can produce. If, however, they ask for figures that are safe and best for normal conditions, this 3 to 5 deg higher condition may not be necessary, and they can take full advantage of the manufacturer's potential design by throttling their steam at part loads and thus maintain their economy.

Again, with the cooling tower on steam jet application, it is possible to use a higher temperature range of cooling tower water, so that your tower size is not comparable on the constant heat release per unit of volume basis. You can release the heat at higher temperature heads and cut down your size, so that instead of having a tower, as I have heard said quite often, three times the size for a steam jet machine, as you do for a compressor machine, towers of identical sizes for the same capacity have been used by taking advantage of the potential higher temperature range through which the cooling may be accomplished in the case of a steam jet, without loss of compression efficiency.

MR. BROWN: Pertinent to this paper is a new venture in imposing a hardship on the air conditioning user regarding his condenser water disposal. This is regarding the disposal only, regardless of whether the user obtains his water from his own well or purchases it from the city. In Columbus, Ohio, they just passed a city ordinance, which is being set in motion at the present time, to charge a sewage disposal tax of 45¢ per thousand cubic feet, which is almost as much as the water cost to begin with, and when you add that to a city water cost as high as we have in some cities, such as New York, you get very expensive condensing water.

Table 1 gives the evaporative loss and the drift loss. I would like to just read some figures I jotted down translating that into terms of a 200-ton plant, because these fractional gallon figures mean very little to most of us. For a 200-ton plant, the table would read something like this: For evaporative loss, free and evaporative loss 5.5 gal; drift loss 6; total 11.5 gal; for steam jet evaporative loss 16; drift loss 17.7; total 33.7. When you get those figures for an actual operating plant, you will notice that the drift loss is given as slightly in excess of the evaporation loss. That is, I believe, an error except for an extremely inferior-designed tower such as a man might build himself, but any that would be purchased on the market today and any certainly that were purchased in conformance to a consulting engineer's specifications would be held to a drift loss of not over two-tenths of 1 per cent of the water flow, rather than 1 per cent, and there are several towers operating on low roof levels within a block of this building that have a drift loss of only one one-hundredths of 1 per cent. That drops that drift loss figure down to where it is negligible and to where the total figures become evaporative loss figures for all practical estimating purposes.

Since that is the case and since towers can be bought to those specifications, an interesting and useful estimating device is available for getting make-up water costs. Having designed the condenser, and knowing the water flow and the range from the heat release, take the water range, for example 12 deg, say one-tenth of that, 1.2, express that as a per cent and multiply that per cent times the water flow, and that will give you a total make-up water figure covering evaporation loss, plus an adequate provision to cover a reasonable spray loss of one-tenth of 1 per cent.

PROF. W. H. SEVERNS: Mr. Rottmayer's discussion has covered many points in connection with cooling towers. I think it would have been desirable if his paper could have included also something in regard to the design and selection of a tower for a given purpose. For example in connection with a wind or open deck tower if the relative proportions of height, width, depth and deck area could be included for a given quantity of water being handled.

S. I. ROTTMAYER: Regarding Mr. Hertzler's comment on the use of a double spray air washer as a water-cooling device as compared to a cooling tower, the counterflow principle of heat transfer, which is built into the ordinary commercial cooling tower equipment, does not exist in the ordinary commercial air washer as we can buy them from the manufacturers, due to the fact that the gravity effect on the water pulls out the water from the air circuit before it has a chance to follow the air all the way through. If stages, as he suggests, for distributing this water from the first bank to the second and so on, were employed, the counterflow principle of heat transfer could be approached in effect. The cost of this apparatus, however, might have an influence on the selection of it, as compared to the cost of commercial cooling towers.

Regarding the comments on the steam jet efficiency at low loads, as compared with the dichlorodifluoromethane system, without taking time or having the opportunity to digest the data and the figures presented, I might state that the illustration of two comparable machines, one a steam jet machine and another a dichlorodifluoromethane system of the same capacity, when compared on an operating basis, would show that the steam jet system, I believe, will have a higher efficiency at the low loads, due to the fact that the frictional resistance is constant throughout the full range of loads from 0 to 100 per cent on a mechanical compressor, whereas with the steam jet compressor this situation does not exist, and for the lighter loads therefore the efficiencies will be higher with the steam jet system.

The influence of the reduction in condenser water due to the lower wet-bulb temperature available on the lighter load conditions has a similar and proportional effect on both types of systems.

The use of a number of units with a mechanical compressor type of system, of course, would vary or influence the selection of equipment from an economic standpoint. A mechanical compressor should be selected if your load is going to be around 25 or 50 per cent of your maximum, whereas with a steam jet system usually the load is taken care of either by cutting in or out more or less steam jets.

Further comment was made by Mr. Shepard on the range of sizes in which an evaporative condenser may be used. I might mention that evaporative condensers have also been used in connection with steam jet equipment. There is an installation in New York, I understand, which has been in successful operation with this combination.

With reference to the question of Professor Kayan concerning Table 2, these data represent costs based on one ton of refrigeration that have been condensed from larger size equipment, reduced to a common denominator of one ton of refrigeration and are based on the various units that I mentioned in the paper.

The efficiency that I used for the cooling towers was 67 per cent, which is the usual commercial cooling tower efficiency. They may be designed for any range of efficiency which may be specified.

Further comment was brought out by Mr. Brown concerning the tax placed on water disposal. I might make mention of the fact that the use of wells for disposing of condenser water has been employed in certain localities where wasting the condenser water into the city equipment was not permitted.

The effect of drift loss on the power required may also influence the design of the tower; that is a tower may be designed to have any drift loss which may be specified, but usually the wind power available will limit the amount of baffling that it is practical to make use of to prevent excessive drift losses.

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AIR DISTRIBUTION FROM SIDE WALL OUTLETS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Wisconsin, conducted under the supervision of the Research Technical Advisory Committee on Air Distribution.

THE selection of outlets, which will deliver sufficient air for effectively cooling a space and accomplish this within the limits of air motion and temperature required for comfort, often presents a difficult problem to the engineer. Although there are other satisfactory methods of supplying air to a room, this paper will discuss only the problems in connection with the introduction of air through side-wall openings.

The Practical Problem

The engineer who selects the air distribution outlets usually has the following information to guide him: (1) The room dimensions are known; (2) A temperature differential between the entering air and the control temperature of the room has been selected; (3) The air volume required for supplying the necessary cooling has been computed; (4) The maximum velocity permissible from the standpoint of noise has been calculated; (5) The desired grille mounting height is known, either exactly or within rather close limits; (6) The desired minimum room temperature is also known.

The problem is to find the smallest outlet, or outlets (bearing in mind the noise limitation on velocity) which will give satisfactory air distribution. The range of selection is often limited and the entire problem complicated by additional considerations such as beamed ceilings, large posts or other obstructions, non-rectangular rooms, and the location of available space for the outlet itself or the ducts leading to it.

It is often possible to find two or more arrangements of size and location of outlets which will be satisfactory. Generally, the one involving the smallest total outlet area is the most satisfactory.

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The Criteria for Comfort

The necessity for some standard of satisfactory air distribution is apparent. The literature[1] on air distribution is meager and, as none of the papers deal primarily with this subject, it was necessary to make some study from which suitable criteria could be established. That work will not be discussed here except to say that the observations were limited and the subjects few and not sufficiently diversified as to types of people. Nevertheless, these observations, together with the data in the literature referred to, supplemented by considerable field experience, were used to establish a standard test procedure to determine whether or not the distribution in a given room under a given set of conditions was or was not satisfactory.

The accompanying chart of Fig. 1 is a family of curves, each curve representing the maximum permissible difference in room temperature for various air velocities. All combinations below the line are considered to be unsatisfactory, and those above the line satisfactory. It will be noted that, with higher room temperatures, a higher velocity is permitted with the same air temperature difference, or a greater temperature differential is permitted with the same velocity. The velocities referred to are average velocities as measured with a Kata thermometer. The temperature difference referred to is in every case the difference between the temperature of the air at the 5-ft level directly below the inlet and the temperature at any location where the air velocity is measured. All temperatures are measured with exposed thermocouples. It will be observed that this chart is in substantial agreement with what data there are on this subject in the literature referred to.

Theoretically, the establishment of the fact that any room meets the requirements as shown by this chart involves the necessity of measuring the air velocity and temperature difference at every point in the occupied zone. However, with experience one can locate the worst places by inspection, combined with approximate readings of temperature and velocity, so that it is rarely necessary to make more than a few accurate measurements.

Since the requirement stated only sets a high limit, it does not take care of the possibility of improperly cooling certain sections of the room. One solution is to measure temperature rise, but, except near places where there is heat gain, this is not satisfactory, for the temperature rise will be negligibly small, yet when the space is in use the occupants may increase the temperature and relative humidity to an extent which may feel uncomfortable. A chart similar to that shown but specifying *minimum* velocities and air temperature differences is required, but again one must measure the air velocity and temperature at all points in the room and must make allowance for the heat gain in that region. In an odd-shaped room it is often difficult to determine by inspection where the worst places will occur. However, in a square or rectangular room with outlets and return grilles on the same wall and without greatly localized heat sources other than occupants, experience shows that it is usually only necessary to know that sufficient cooling air will reach the

[1] Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker, and C. Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145).
A.S.H.V.E. GUIDE, 1937, Chapter 3, p. 76.
Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H. H. Trimble, C. Gutberlet, and M. F. Lichtenfels (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, pp. 268-278).

far end of the room. The air discharged into the room, in finding its way back to the return grille, necessarily passes below the entering air. The large cross-section available for the return air insures that the velocities are low, and the symmetry of the rectangular room insures that the cooling air will be well distributed.

Even for this simple room there is no single value for the velocity at the rear wall which will satisfy all cases. For example, if the rear wall is a large source of heat gain, the velocity down it should be considerably greater than if it is thoroughly insulated and therefore a source of little or no heat gain. Furthermore, the proper velocity depends to some extent upon the type of occupancy. However, experience indicates that an average velocity of 30 fpm is about the minimum that can be permitted down the back wall. A

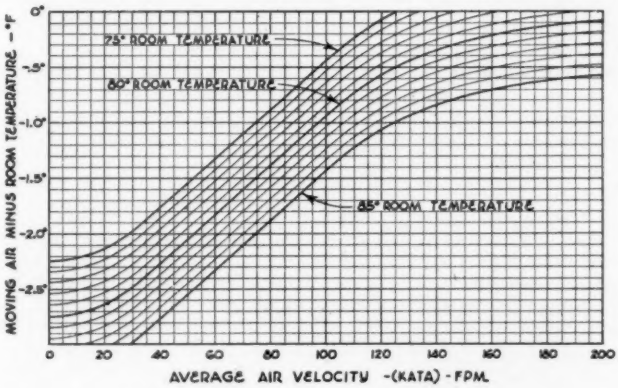

FIG. 1. RELATION BETWEEN MAXIMUM AIR VELOCITY AND MOVING AIR TEMPERATURE FOR VARIOUS ROOM TEMPERATURES

slightly lower value will be satisfactory if there will be no heat gain from that wall.

The maximum permissible velocity down the back wall is controlled by considerations of the discomfort produced upon occupants of the room seated near that wall. The proper method of determining this maximum velocity would be in accordance with the chart, Fig. 1. The temperature difference of the air falling down the back wall will, in the case of most outlets which are otherwise satisfactory, rarely exceed 1 F. Accordingly, the maximum permissible velocity, if the room temperature were 80, would be 100 fpm; and if the room temperature were 75, would be 75 fpm. It is well to keep this particular velocity considerably on the low side, since air which comes down the back wall is deflected at the floor and returns along the floor where the occupant is particularly sensitive to a feeling of discomfort from a low temperature. Accordingly, an arbitrary requirement was made that the average velocity down the back wall should not exceed 75 fpm.

There is one other uncomfortable condition not mentioned thus far and not covered by Fig. 1. Regardless of the air temperature, there is some air

velocity, even if momentary, which is uncomfortable if only because it disturbs papers and other light articles or the occupant's hair. This maximum acceptable velocity is believed to be in the neighborhood of 200 fpm, and accordingly one requirement for a satisfactory outlet is that the maximum air velocity within the zone of occupancy shall not be more than 200 fpm as measured with an instrument which indicates approximately the momentary velocity.

The Test Room and Instruments

The room in which the data for this article were taken was constructed particularly for air distribution tests. Fig. 2 is a photograph taken within this room, and Fig. 3 is a diagrammatic plan. It has been partially described in a paper on grille noise,[2] but the description is repeated here to obviate the necessity of referring to the other paper. The building itself is of poured concrete, pillar and slab construction, with a flat concrete ceiling covered with built-up composition roofing. The test room is on the sixth (top) floor of this building, and is 17 ft wide and 9 ft 7 in. high. The inner side walls are of sheet metal, outside of which are narrow spaces, heated to maintain room temperature during tests with cool air. The ceiling is ¼ in. rigid insulation on joists about a foot below the concrete slab of the roof. The floor is concrete. An exhaust grille, extra large to reduce velocity to a negligible value, is located in the center of one wall at the floor. This wall is also of ¼ in. rigid insulation and forms the front of a pressure box 5 ft deep and 8 ft wide extending from the top of the exhaust grille to the ceiling. Air is introduced into this box near the bottom and at velocities greatly reduced by a diverging nozzle with splitters. The front of the pressure box is in sections so arranged that a grille of any size and shape can be installed at any height desired. Behind the test grille is mounted a 3-ft section of duct having the nominal dimension of the grille to simulate an actual installation. Careful tests have indicated that the flow of air from the pressure box through this duct is satisfactorily uniform over its entire cross-section.

The room is heated by a panel system and the room temperature is maintained constant by a control system governing the temperature of the air behind the panel, supplemented with a pilot thermostat in the room. Since very small temperature differences are important in determining whether or not the particular grille meets the velocity and temperature requirements, it is necessary that the room temperature be held within very close limits during the period of one test. A room temperature change of as much as 0.5 F during one set of readings necessitates repetition of the test.

The temperature of the entering air does not need to be so accurately controlled. A temperature variation of as much as 1 F during a single test will not appreciably affect the readings. A standard proportioning control is used in the blast system supplying the air. Frequent thermocouple temperature readings are taken of return and entering air temperatures.

Thermocouples are installed about the room on 5 ft centers horizontally and 2½ ft centers vertically, as indicated in Fig. 3. A number of these are also shown in the photograph of Fig. 2.

[2] The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 81).

The temperature differences were measured directly by the expedient of locating the return junction of the thermocouple galvanometer system in the aspirated air. The return junction thermocouple location actually used for all the readings shown was at the 5 ft level about 1 ft from the wall and directly beneath the air outlet.

The measuring circuit is arranged so that a sensitive galvanometer determines the balancing potential necessary for zero current in the thermocouple circuit, and a potential multiplying system enables the net voltage of the two junctions to be read directly. This voltage is a measure of the difference in temperature at the two points and the voltmeter can be calibrated directly in degrees Fahrenheit. By this means temperature differences were measured

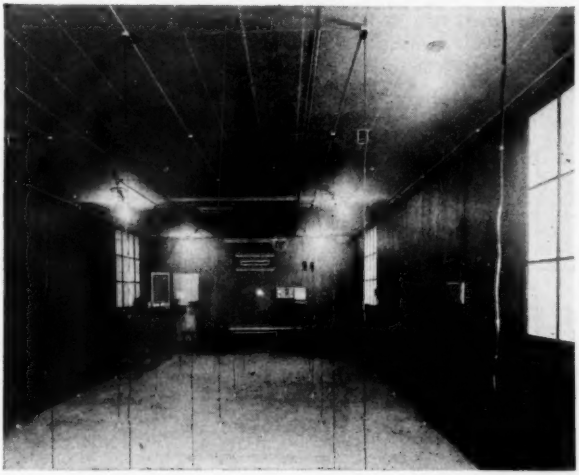

FIG. 2. THE TEST ROOM LOOKING TOWARD THE AIR DISCHARGE OUTLET

with an accuracy of approximately $\pm$ 0.1 F between readings. The absolute value of the differences is not as accurately known as this, but the error, if any, is also in the zero temperature line of Fig. 1.

A recording thermometer of a type having a current of air over the thermostatic element was used to record the room temperature at a point adjacent to the return junction of the thermocouple. Records of room temperature refer to the readings of this instrument. Measurements of the inlet and exhaust air temperatures were made with thermocouples and a commercial potentiometer having its return junction compensated for the ambient temperature. This instrument, as well as the fan recorder, has an accuracy of approximately $\pm$ 0.5 F.

The volume of air discharged was measured with a Wahlen gage and a carefully calibrated Venturi meter located in the duct supplying the pressure box. The leakage from the pressure box at various differences in pressure between it and the room was obtained by measurement of the flow through

the Venturi meter with the grille opening closed. A slope gage indicates the static pressure in the box and the leakage calculated from this pressure was deducted from all Venturi meter readings in determining the total volume discharged into the room. The outlet face velocity was obtained by dividing the volume as determined by the Venturi meter by the *core area* of the grille or opening. By *core area* is meant the actual opening inside of the grille frame in cases where a grille was used, and the area of the opening where no grille was used. As a rough check against the possibility of a gross error in calculation, the face velocity was in each instance measured with a deflecting vane type indicating instrument.

Room velocities were measured either with the deflecting vane type indicating instrument or with a Kata thermometer. High temperature Kata thermometers were used, two or more consecutive readings being averaged for each record. The air temperature was in every case measured by a thermocouple at a point adjacent to the Kata thermometer and at the time the Kata reading was taken. The velocities shown on the charts with this article were

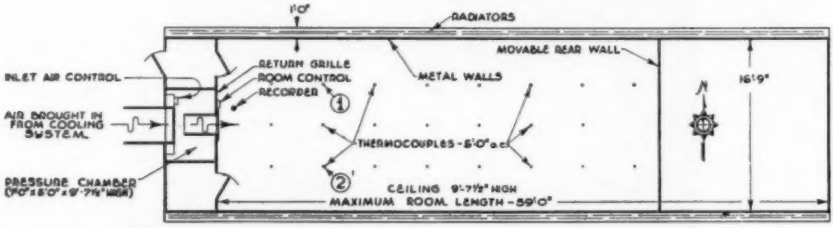

FIG. 3. DIAGRAMMATIC PLAN OF TEST ROOM AND PRESSURE BOX

taken with a deflecting vane instrument and represent mental averages in one direction (what might be called the vector average)—not the non-directional (or scalar) average as measured by the Kata thermometer.

In order to provide for rooms of various lengths, the rear wall of the room is movable, as shown in Fig. 3. The thermocouple drops are inserted into plugs in the ceiling so that they can be removed when the length of the room is to be altered.

The Test Data

The results of one series of tests made in this room are shown in Figs. 4 to 8, inclusive. In this series used for illustrative purposes, five open end ducts and five grilles to fit them were used. The aspect ratio of the grilles varied from 1.00 to 12.2, with the areas maintained fairly constant. All grilles were of one type, with thin horizontal closely spaced guide vanes giving a slight upward deflection to the air stream. In the particular series shown, an entrance velocity of 1000 fpm and a 20 F temperature difference were used throughout; the length of room was fixed at 40 ft, and the grille mounting height was 7 ft 6 in. Each of the figures has two parts, one marked *A* for the test with the open end duct and the other *B* for the test with the grille fitted in place.

Temperature differences were measured with the thermocouples at 120 positions at 5 ft spacings and at five elevations. The 40 readings taken along the vertical center plane of the room in line with the grille are shown on Figs. 4 to 8 as differences between the observed temperature and the room temperature. The edges of the primary stream were first located and marked at each 5 ft location by the use of cigarette smoke, and checked by observing the path of a large quantity of smoke introduced with the air supply. This air stream path is shown on the figures as a broken line. In the air stream the average velocity was determined at elevation intervals of a few inches at each of the 5 ft locations. At each one of 35 thermocouple locations, the average and maximum velocities were taken. In all, about 125 such velocity

TABLE 1—PERFORMANCE CHARACTERISTICS FOR AIR DISCHARGED FROM AN OPEN HOLE
(*Room Length*, 40 *ft; Face Velocity*, 1000 *fpm*)

1		Nominal Size	11 x 11	16 x 8	19 x 7	28 x 5	36 x 4
2		Aspect Ratio	1.00	2.03	2.78	5.84	9.54
3		Frame Area (sq ft)	0.803	0.847	0.879	0.915	0.920
4		Frame Area/Duct Area	0.956	0.954	0.950	0.940	0.920
5		Supply Volume (cfm)	803	847	879	915	920
6		Room Temperature, deg F	80	80	79	81	82
7		Supply Temp. Diff., deg F	− 20	− 20	− 19	− 20	− 20
8	Data at 5 ft level	Max. Peak Velocity[a], fpm	300	250	250	300	350
9		and Location, feet	20	15	20	20	15
10		Av. Vel. Same Loc.[a], fpm	225	200	200	200	300
11		Max. Vel. on CL (Kata), fpm	130	170	142	245	208
12		Location, feet	25	20	20	19	20
13		Temp. Diff., deg F	− 2.5	− 3.3	− 3.1	− 4.0	− 3.4
14		Velocity at (1)[b] (Kata), fpm	40	40	20	25	45
15		Temp. Diff., deg F	− 0.5	+ 1.0	+ 1.5	0	− 0.5
16		Velocity at (2)[b] (Kata), fpm	25	20	38	20	30
17		Temp. Diff., deg F	− 0.5	+ 1.0	+ 1.5	0	0
18		Vel. at Rear Wall (Kata), fpm	100 up	60 up	75 up	90 up	88 up
19		Temp. Diff., deg F	− 1.7	− 1.0	− 0.8	− 2.5	− 0.9
20		Distance from Opening, feet Stream Hits Floor	28	25	27	23	28
21		Distance from Opening, feet Primary Jet Drops to 150 fpm	25	26	26	27	23

[a] Data with deflecting vane type velocity meter.
[b] See Fig. 3 for location.

readings were made in each test. For this purpose a deflecting vane type meter was used and the determinations of the average velocities were used to plot lines of equal velocities on the figures. In addition, Kata thermometer readings were made at certain points in the room. Some of the pertinent details of these tests are summarized in Tables 1 and 2.

Lines of equal velocity were plotted on Figs. 4 to 8, using the large number of readings obtained by the deflecting vane instrument, which made it possible to trace out the major air movements in the room. Naturally, there were many miscellaneous movements or turbulences that can hardly be recorded or plotted. The averages were obtained by observing the swing of the indicating hand of the instrument. Since this type of instrument is directional, it is necessary to hold it into the direction from which the air is coming. This is

difficult to do under turbulent conditions and the visual averaging of the swing of the hand also adds to the difficulty of obtaining much accuracy in these determinations. Four different instruments of this type were available and were found to check very closely among themselves, and are considered to be satisfactory for tracing the air currents as plotted on these charts. It should be borne in mind that the values shown in Figs. 4 to 8 are directional and should not be used in interpreting comfort conditions in connection with Fig. 1. The velocities shown on Fig. 1 are non-directional as determined by a Kata thermometer. Velocity readings were taken at various points in the

TABLE 2—PERFORMANCE CHARACTERISTICS FOR AIR DISCHARGED FROM A GRILLE
(*Room Length*, 40 *ft*; *Face Velocity*, 1000 *fpm*)

1		Nominal Size	11 x 11	16 x 8	19 x 7	28 x 5	36 x 4
2		Aspect Ratio	1.00	2.17	3.04	6.94	12.2
3		Core Area (sq ft)	0.677	0.711	0.729	0.723	0.696
4		Core Area/Duct Area	0.806	0.800	0.790	0.744	0.696
5		Supply Volume (cfm)	677	711	729	723	696
6		Room Temperature, deg F	81	80	81	80.5	81.5
7		Supply Temp. Diff., deg F	− 20.5	− 20	− 21	− 21	− 20
8	Data at 5 ft level	Max. Peak Velocity[a], fpm	185	180	180	225	230
9		and Location, feet	25	30	30	25	25
10		Av. Vel. Same Loc.[a], fpm	125	75	100	140	100
11		Max. Vel. on CL (Kata), fpm	70	70	55	150	120
12		Location, feet	28	30	30	23	21
13		Temp. Diff., deg F	− 0.7	− 1.5	− 1.3	− 3.0	− 1.3
14		Velocity at (1)[b] (Kata), fpm	28	24	20	34	15
15		Temp. Diff., deg F	0	+ 1.0	− 0.5	+ 0.5	− 0.5
16		Velocity at (2)[b] (Kata), fpm	20	20	10	36	25
17		Temp. Diff., deg F	0	+ 1.0	− 0.5	+ 0.5	− 0.7
18		Vel. at Rear Wall (Kata), fpm	120 dn.	120 dn.	55 dn.	50 tur.	55 tur.
19		Temp. Diff., deg F	− 2.1	− 1.6	− 1.2	− 1.8	− 1.0
20		Distance from Grille, feet. Stream Hits Floor	40 +	40 +	37[c]	40[c]	40[c]
21		Distance from Grille, feet. Primary Jet Drops to 150 fpm	40 +	40 +	35	34	28

[a] Data with deflecting vane type velocity meter.
[b] See Fig. 3 for location.
[c] Actual stream diffuses before reaching floor. Distance given is where lower stream edge extended would strike.

room with Kata thermometers as recorded in Tables 1 and 2. The four Kata instruments available gave consistent results.

The temperature differences obtained in the center plane of the room are shown on the charts on the 5 ft vertical lines in the locations in which they were observed. For clarity, the velocity determinations are placed on the intermediate vertical lines. The method used in taking and in plotting velocities and temperature differences is considered to result in a good picture of the flow pattern of the various openings and the related temperature differences.

The items in Tables 1 and 2 are perhaps self-explanatory, although attention might be called to a few of them. For the open hole tests the maximum peak velocities were found to be from 250 to 350 fpm at a distance 15 to 20 ft from

the supply outlet. The average velocity at the same locations was found to be 200 to 300 fpm. In the case of the grille tests, the maximum peak velocities varied from 180 to 230 fpm at a distance of 25 to 30 ft out, and the average velocities were 75 to 140 fpm at the same location. These figures are given

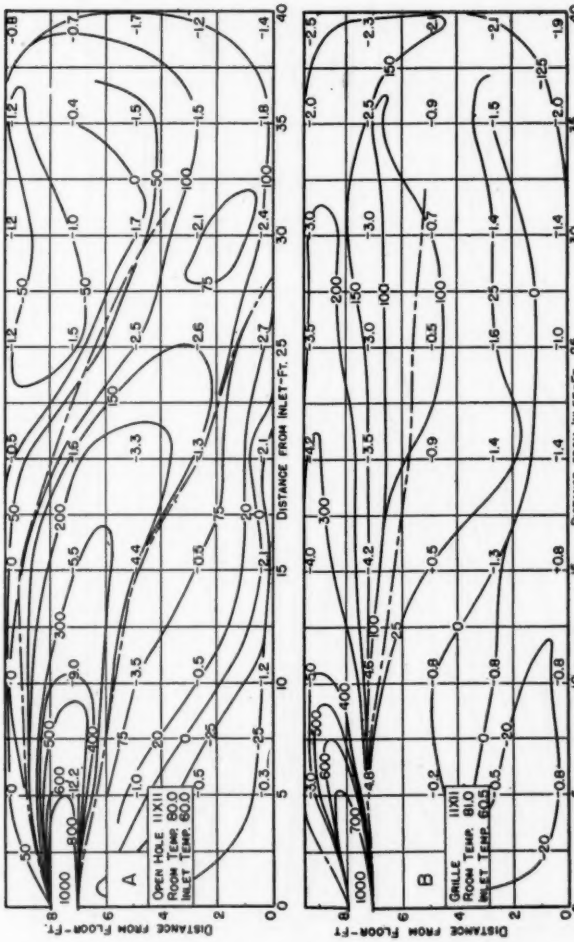

FIG. 4. AIR DISTRIBUTION ALONG VERTICAL SECTION THROUGH CENTER OF ROOM: *A (above)* FOR 11 x 11 OPEN HOLE; AND *B (below)* FOR 11 x 11 GRILLE

in items 8, 9 and 10 of Tables 1 and 2. These particular figures do not clearly indicate any effect due to aspect ratio. As pointed out elsewhere, the velocities measured with an indicating vane meter are useful for the purpose of obtaining a general picture of the flow pattern, but are not accurate when the stream is very turbulent.

Items 11 to 19 of these tables concern comfort conditions and may be used in connection with standards such as shown in Fig. 1 to indicate satisfactory or unsatisfactory performance. The highest velocities, as determined by the Kata thermometer, and temperature differences found at the 5 ft level indicate

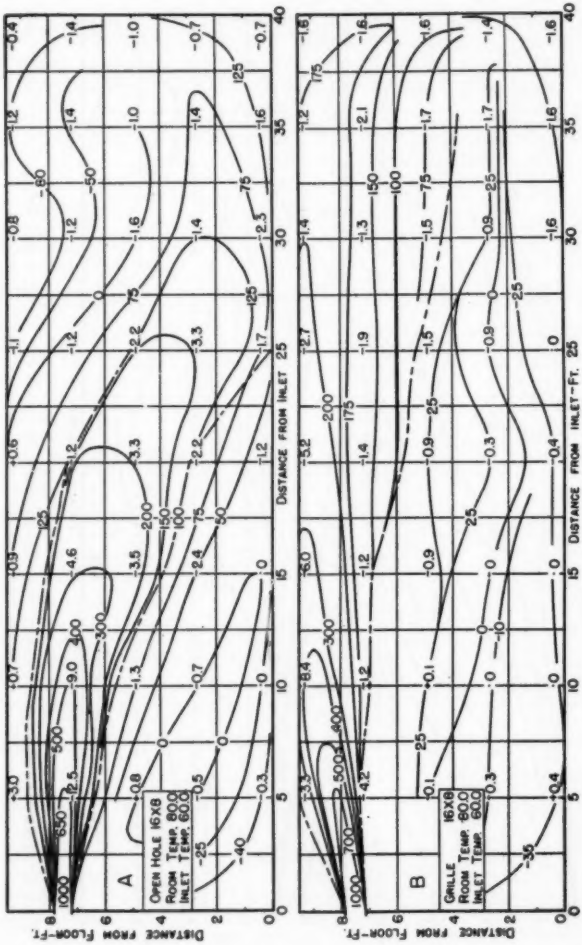

FIG. 5. AIR DISTRIBUTION ALONG VERTICAL SECTION THROUGH CENTER OF ROOM: *A (above)* FOR 16 x 8 OPEN HOLE; AND *B (below)* FOR 16 x 8 GRILLE

that unsatisfactory combinations existed in all of the open hole tests. These velocities range from 130 to 250 fpm, and the temperature differences from — 2.5 F to — 4 F. In the grille tests the three with low aspect ratios showed satisfactory performance at all distances at the 5 ft level, as judged by standards of Fig. 1. The two with high aspect ratios showed higher velocities,

with greater temperature differences at locations nearer the supply outlet than did the other three grilles. Reference to Fig. 1 indicates unsatisfactory performance for these two. An increase in outlet velocity might remedy this unsatisfactory performance, but without an increased length of room might

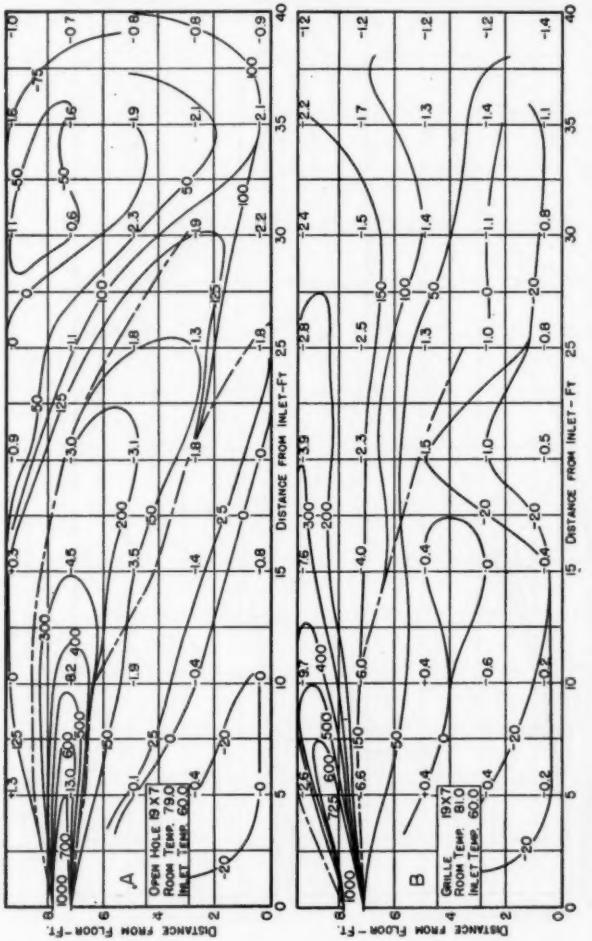

FIG. 6. AIR DISTRIBUTION ALONG VERTICAL SECTION THROUGH CENTER OF ROOM: *A (above)* FOR 19 x 7 OPEN HOLE; AND *B (below)* FOR 19 x 7 GRILLE

result in an excessive velocity down the rear wall. The maximum Kata thermometer velocities for all grilles range from 55 to 150 fpm, and the temperature differences from — 0.7 F to — 3 F.

The air movement in the return air stream flowing towards the return grille is indicated in items 14 to 17. For the open hole tests, the average velocity

at points 1 and 2 on Fig. 3 was 30 fpm, and the average temperature difference was between $+0.3$ and $+0.4$ F. For the grille test, the corresponding average velocity was 23 fpm at an average temperature difference near $+0.1$ F. The difference in velocities is about in the same proportion as the ratio of

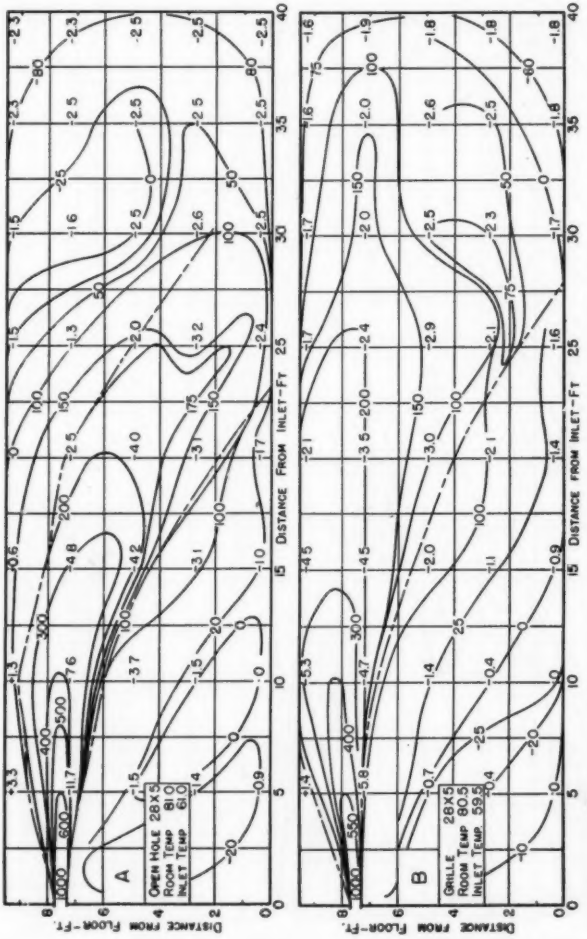

FIG. 7. AIR DISTRIBUTION ALONG VERTICAL SECTION THROUGH CENTER OF ROOM: *A (above)* FOR 28 X 5 OPEN HOLE; AND *B (below)* FOR 28 X 5 GRILLE

average volumes in item 5 of the two tables. The slight difference in temperature differences is probably of no significance.

In each of the open hole tests, the velocity at the back wall is upward and, as far as the standards of Fig. 1 are concerned, is on the border line of acceptability, but actually is not to be tolerated, as the upward flow indicates

falling short. For the first two grilles tested, the flow is definitely down and the velocity somewhat beyond the acceptable limit as indicated in Fig. 1. The air stream is therefore reaching the rear wall at too high a velocity. The performance probably would be satisfactory in a slightly longer room.

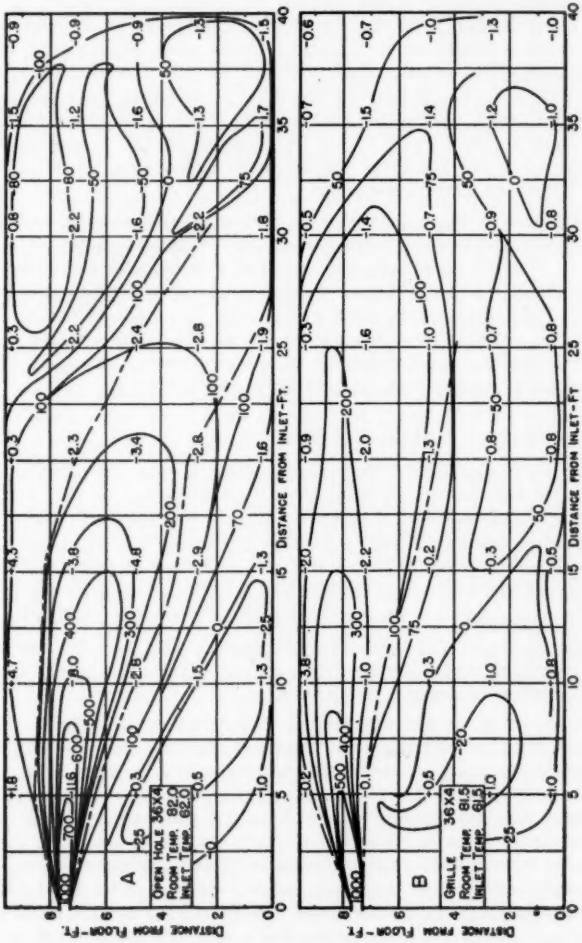

FIG. 8. AIR DISTRIBUTION ALONG VERTICAL SECTION THROUGH CENTER OF ROOM: *A (above)* FOR 36 x 4 OPEN HOLE; AND *B (below)* FOR 36 x 4 GRILLE

For the 40 ft room a slightly lower velocity probably would result in satisfactory performance. The third grille, 19 x 7, had an acceptable velocity down the rear wall and, in fact, is acceptable in every respect in accordance with the requirements set forth in this paper. The two remaining grilles with high aspect ratios had low velocities at the rear wall. These velocities were

rather mixed, although apparently more upward than downward. These two grilles, however, as shown in item 11, have excessive velocities at the 5 ft level.

Items 20 and 21 showing velocities taken with the deflecting vane type meter indicate the length of throw of the various air streams. With the open hole, little or no effect is shown on the throw with change in aspect ratio. With the grilles, some tendency to a shortened throw with high aspect ratio is indicated, although this is somewhat obscured by the influence of diffusion. In the open hole test, the stream hit the floor at distances from 23 to 28 ft from the supply outlet. When the grilles were in place, because of diffusion, the stream could hardly be said to hit the floor at all, but a continuation of the lower edge of it for the three higher aspect ratios would strike the floor between 37 ft and the end of the room.

The location where the air stream is reduced to an average velocity of 150 fpm, as indicated by a deflecting vane type meter, is shown in item 21 of the tables. For the open hole tests, no influence is indicated of the effect of the aspect ratio. The air stream reaches the rear wall with this velocity in the tests of the low aspect ratio grilles. In the grille with the highest aspect ratio this velocity was reached at 28 ft out. The average distance measured horizontally for the reduction of the air stream velocity to 150 fpm was 25 ft for the open hole tests and 36 ft for the grille tests.

Factors Affecting the Distribution of Air

A quantitative discussion of the proper outlet to use for even a limited number of representative rooms is much beyond the scope of this paper, but it is possible to point out in a qualitative manner the effects of some of the significant factors involved. The following statements are made as a result of a large number of experiments of the sort described in this paper, supplemented by experience with actual installations. A list of some of the principal variables other than throw (usually the length of the room) follows:

(1) Height of room; (2) Mounting height of grille; (3) Volume of air; (4) Temperature of air; (5) Velocity of air; (6) Aspect ratio; (7) Side walls; (8) Location of outlet; (9) Beams.

The first two of these are interrelated to a considerable extent. For example, in a high room with the grille mounted near the ceiling, there will be little or no difficulty due to cold air dropping into the occupancy zone, regardless of all other considerations. There remains, however, the problem of making certain that the cooling reaches the far end, although the greater mounting height tends to increase the throw. On the other hand, for room heights usually encountered (except in theaters and auditoriums) the throw is not materially affected by the room height. If the outlet is close to the ceiling, the lessened friction at the upper side of the air stream tends to hold the cold air up. However, if the grille has a considerable upward deflection and is located adjacent to the ceiling, there will be a tendency for the air to rebound from the ceiling and drop into the occupancy zone. It is apparent that the proper type of grille is much affected by the distance between the grille and the ceiling.

As might be expected, a large volume of air has more tendency to drop

into the occupied zone than a small volume of air. The larger volume will have a longer throw, all other things being equal. As previously pointed out, a room may be unsatisfactory because too much cold air drops down into the occupancy zone or because it strikes the rear wall and follows that down into the occupancy zone.

As one would also expect, the greater the difference in temperature between that of the room and the entering air, the more tendency there is for the entering air to fall into the occupancy zone.

The effect of the velocity of the entering air is not so simple and is, in fact, very complicated with certain types of grilles. For an open hole, an increase in air velocity will cause the point at which the cool air first drops into the occupancy zone to move toward the rear wall, but may increase the velocity of this objectionable current without appreciably reducing its temperature. The increased entering velocity may raise the velocity down the rear wall to an excessive figure. The effect of velocity upon the flow characteristics from any particular grille can only be determined by a rather elaborate test procedure simulating the conditions of actual use.

Size of the grille is a factor, but, as it is fixed by the volume of air required and the velocity, it does not need to be discussed or considered separately.

Aspect ratio is the width divided by the height of the actual opening inside the supporting frame. It is a factor in distribution, but except when quite large is not as important a factor as appears to be the general opinion. Nevertheless, a high aspect ratio does decrease the throw. If the grille is mounted near the ceiling, the high aspect ratio will increase the rate at which the cool air mixes with the room air and therefore decrease its tendency to fall. If the grille is mounted at a considerable distance below the ceiling (for example, 2 or 3 ft) and is of the upward deflecting type, the effect just referred to will be overcome in many instances by the effect of the high aspect ratio in preventing the air stream from being carried high enough to slide along the ceiling. An example of this combination of circumstances will be seen in the charts of Figs. 7-*B* and 8-*B*.

The practical problem of selecting a suitable grille is often greatly complicated by the width of the room and the necessity of locating the outlets at particular places. For example, a wide room requires either a grille with which the air can be forced to diverge widely from the face of the grille, or a number of outlets must be used. Grilles which give a wide divergence are not suitable for very long throws, and this fact must be kept in mind by the designer. Narrow rooms necessitate the use of a grille which will not give a wide divergence and, generally speaking, such a grille will produce a relatively long throw. A widely divergent grille in a narrow room may result in high velocity down the side wall with consequent discomfort to the occupants near the wall. For architectural reasons it is often necessary to locate the outlet at or near a corner, and for this purpose some sort of outlet must be used which will direct the air away from the adjacent wall. Beams are often encountered. If parallel to the direction of air flow, they are not ordinarily disturbing, although this is not true if a wide angle grille is necessitated by other considerations. If the beams are at right angles to the only direction from which it is possible to discharge the air, the designer must be sure that the entering air does not strike against the beam, for it will be deflected down into the occupancy zone.

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CONCLUSIONS

There is a need for an authoritative set of requirements for determining comfortable conditions within a cooled space. Criteria are needed to emphasize that comfort air conditioning specifications should include a relation between the three factors: room temperature, air velocity, and the difference between the room temperature and that of the moving air.

The data published in this paper indicate that for the conditions tested a plain side wall opening is unsatisfactory. More tests and much experience have shown it to be unsatisfactory in most types of rooms. Other tests and experience have likewise demonstrated that nearly all actual installation problems can be solved by use of some type of grille now available.

ACKNOWLEDGMENT

The Research Technical Advisory Committee on Air Distribution is indebted to the Barber-Colman Co. for providing the facilities of their laboratory and the necessary personnel to assist in performing the tests and correlating the data.

DISCUSSION

PROF. B. M. WOODS: These problems of velocities, combined with temperature differences are, of course, among the most serious problems before us. I am grateful to Professor Nelson for illustrating the high variability of flow under these different conditions. It is important to know something more about the characteristics of the grille.

We have conducted some interesting experiments in the laboratory at the University of California indicating the very slow progress of diffusion. Results are so contrary to our common thinking that I think it is worth mentioning just one or two. If you discharge natural gas in the air in order to get two gases of different densities, natural gas being about two-thirds as heavy as air, at a sufficiently low velocity from a nozzle to have approximately laminar flow, you can obtain pictures by differential densities in the form of shadow-graphs. If you will do that, even though you cannot see the conditions, you can presently picture them due to the differential densities. By using a nozzle discharge, not very large, about $\frac{1}{16}$-in. in diameter, and a flow in the laminar region, it is possible to get a flow of the gas through the air for 3 ft before it spreads to the width of a pencil.

When you think about diffusion and mixing, you get a false notion if you consider it a rapid process. It is a very slow process, and mechanical mixing, of course, is the important thing. Next in importance is to get above the Reynold's Number for laminar flow and thus get into turbulent motion. When that is done you get a definite conical angle of some degree and a violent mechanical mixing. That is what we are after in air distribution, and so it seems to me quite important that we should have adequate experiments to determine how soon and under what conditions the air which is being discharged reaches this turbulent state. As soon as mechanical mixtures of fairly wide conical angles are obtained the temperature difference between the cool air and the air below will be reduced, and higher velocities for distribution, and therefore lower capacities of fans will be permissible.

D. J. STEWART: Professor Woods asked what style grille was used. It was a horizontal fin spaced $\frac{3}{16}$-in. with each fin having an upward deflection of approxi-

mately 5 deg. That is, the discharge from the face of the grille is approximately 5 deg upward. He also discussed the problem of turbulence. The turbulence at the face of a grille of this sort is very great as far as one can determine from observation of the smoke. Another fact which indicates that the turbulence is great is the rapidity with which the temperature of the air increases. I think one of the charts showed a figure of 4.8 F below the room temperature at a distance of about 5 ft from the face of the grille. This is a drop, or rather a rise of approximately 15 deg in that short distance. Some of this rise may be due to conduction, but that probably is a minor factor. Most of the difference in temperature is due to turbulence.

PROF. G. L. TUVE: We have not given due importance to this idea. The public evaluates air conditions by what happens inside the room, and this matter of air distribution is, therefore, very important. It seems to me that there are two problems here; one of them is what the grille will do when it is out in the open; the other is what it will do when it impinges against ceilings, backwalls or sidewalls.

As I understand the paper and the graphs given, most of the so-called satisfactory performance involved ceiling impacts with the skidding of the air along the ceiling toward the back wall. There is some question as to whether these values are strictly comparable with the performance of the plain opening which did not have the advantage of that skidding action, which will carry the jet considerably farther along the line of travel.

The matter of air measurement is in a very precarious state with regard to this entire question. How do you measure the velocity of air? The authors of the paper used two methods, and there was not any very good interconnection.

But I want to emphasize especially this difference between the two kinds of performance, the kind of performance in the open room, where you would not obtain nearly as long a throw, and the kind of performance where you allow the air to impinge against the ceiling, and you get an entirely different distribution in the room.

I should like to ask one question regarding the temperature difference, as to whether the temperature difference was based on a spot reading at one point as compared with the reading of the temperature in the grille.

MR. STEWART: With respect to the difference between a set of measurements made with the air introduced from the side walls and a set of measurements made with the air introduced from some point out in the body of the room, the question of introducing the air into the center of the room is, as Professor Tuve says, an entirely different problem. These tests do not have any bearing on that problem at all. However, the practical problem is usually that of introducing the air from a sidewall.

The next point concerns the distance between the ceiling and the discharge outlet. In the tests which were presented, that distance was approximately 2 ft. That distance has a very important bearing upon the distribution and upon the type of grille which will be suitable in any given set of conditions. This is especially so when one has a distance between the ceiling and the grille of less than 2 ft. A materially different type of grille is satisfactory, for example, at 1 ft than at 2 ft. When the distance becomes greater than perhaps 3 or 4 ft, additional great distances do not make much difference.

As to the method of measurement, it is our opinion that the Kata thermometer is the best instrument that we know of at the present time. Tests of the sort that were indicated here with the vane meter were for the purpose of giving a picture of the flow patterns. To have done the same thing, to have made the very great number of tests that were required to get that flow pattern and make them with the Kata thermometer, would have taken two or three men many months. Actually no such large amount of measurement is necessary to determine whether or not the room will be satisfactory. An experienced man soon learns to tell reasonably well

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where the worst places will be, and four or five determinations will suffice to check a room under any given set of conditions.

The temperature differences Professor Tuve mentioned I did not quite understand, but I may say that the figures shown and those regularly used are measured by measuring the difference in electromotive force from two thermocouples, one located at the point to be measured and one located at that point near the return grille where the recording thermometer was shown in the photograph.

PROF. F. B. ROWLEY: It seems to me that the importance of air distribution as a part of any air conditioning system is often under-estimated. I think that it has been aptly said that an air conditioning system is no better than the distribution system. Probably more failures can be attributed to poor distribution than to any other phase of the system, and a large part of these defects can be traced to lack of knowledge as to how to design a good system. I feel that the authors are to be congratulated on an exceptionally fine piece of work and hope that as further reports are made more details may be added as to the grille construction. This represents a type of cooperative research work that is profitable to the Society and I think we are indebted to the University of Wisconsin and also to the Barber-Colman Co. who have cooperated with the Research Committee of the Society in making the work possible. There ought to be more such cooperative projects.

PROF. J. E. NICHOLAS:[3] There is no doubt, I am sure, in anyone's mind how very important it is to know something about diffusion of the air. Having built a house recently, I appreciate how much more I would have liked to have known about air diffusion before I started. May I ask if the incoming air was colder than the inside air in which the diffusion was measured, and what would have been the effect on the diffusion? Under the experimental set up that means, of course, that the momentum of the incoming cold air per cubic feet at the same velocity was greater and that no doubt helped in diffusion. Have the authors considered the importance of measuring their relative humidity?

MR. STEWART: The importance of measuring the relative humidity in establishing the chart, which was the first figure in the paper, is great. Actually, that chart was established with a relative humidity of approximately 40 per cent. I would like to make a few remarks to clear up any idea in any one's mind that the authors believe this chart of the first figure to be the last word on that subject. In the first place, our subjects were rather few and the number of observations limited. We had to have something for a criterion. Since this paper was written, I have seen Mr. Houghten's paper.[4] Those of you who have read it will notice that he has done a very thorough job of just this thing, but at a lower room temperature, 70 deg. It isn't safe to say at all surely, but it does look as though the same tests repeated at the higher room temperatures would not fall far from the curves of Fig. 1. Those curves of Fig. 1 ought to show, of course, the relative humidity and there ought to be several sets of them for different relative humidities.

[3] Professor, Dept. of Agric. Engrg., Pennsylvania State College, State College, Pa.

[4] Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth, by F. C. Houghten, Carl Gutberlet and Edward Witkowski (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 289).

No. 1077

CONDENSATION WITHIN WALLS

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MINNEAPOLIS, MINN.

THE condensation of moisture or formation of frost in the various parts of buildings in which low temperatures and high humidities occur has always produced a troublesome problem. It is caused by a combination of conditions, and in cold climates may be expected to a greater or lesser extent in any public building, private home, industrial building such as a creamery or laundry, or any other structure in which high humidities are carried. It may be found in cold storage buildings located in any climate. The problem is not new, but in the past has been accepted as inevitable under certain conditions and has been reasonably well solved by using special construction and other precautions in those places where trouble was apt to occur. Recently there have been marked advances in the science of air conditioning and building construction. Changes have been made which have completely upset the satisfactory application of the old and accepted standards to the newer types of buildings and living conditions. Many of our residences and public buildings which are located in colder climates are now in the danger zone, whereas under former conditions they would have been entirely free from the possibilities of condensation. The magnitude of these new moisture problems has caused some people to doubt the real value of the improvements. Others who accept the benefits of the improved conditions are inclined to shift the responsibility for any difficulties encountered to one of the several causes in which they are not specifically interested. Thus to some the entire trouble is due to air conditioning, to others it is insulation, and to still others it is due to the application of weatherstrips and other methods of providing tighter building construction.

The condensation of moisture takes place when water vapor is cooled below its saturation temperature. In the cases under consideration air is also present with the vapor, but the presence of air does not change the condensing temperatures of the vapor. This condensation of moisture may occur on the surface of a wall or at any place within the structure where the temperature of the material is below the saturation or dew-point temperature of the vapor. If these temperatures are below the freezing point then the condensation will result in an accumulation of frost and ice. The temperatures of the surfaces

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and interior parts of the wall may be calculated with reasonable accuracy from the known thermal properties of the materials used in the walls. The dew-point temperature of the air to be carried within a room may be obtained from any psychrometric chart and thus the danger point of any interior wall surfaces may be anticipated. The dew-point temperature of the air within the walls may also be determined from a psychrometric chart providing its conditions are known. Unfortunately the vapor transmitting properties of building materials are not so well understood as the heat transmitting properties are and the moisture conditions within a wall cannot be calculated so readily. An extended study of this phase of the problem is under investigation at the Forest Products Laboratory, Madison, Wis., and a recently published paper [1] contains some valuable information and recommendations concerning the moisture problem. The author has made an excellent analysis of the problem and has shown by diagrams how insulation will reduce the temperatures of the outside sections of a wall and may thereby increase the possibilities of trouble in these areas.

The fundamental causes of condensation problems are high humidities and low temperatures of the materials with which the vapor comes in contact. Several of the changes which have recently taken place in air conditioning requirements and building construction have greatly increased the probability of meeting these conditions. These changes are: first, raising the relative humidity within the structure by artificial means, second, the application of weatherstrips and other means to prevent the filtration of air through the walls of the building, and third, the addition of better insulation in the walls of the building. These three factors are all necessary if the benefits of modern scientific developments are to be enjoyed and all have contributed to the moisture problem. It cannot be solved by placing the responsibility on any single factor.

For many years the low relative humidity found in the average residence and public building located in cold climates has been a subject of much discussion and the effect of these low humidities on both health and furnishings has been controversial. The net result has been an increase in the average relative humidities carried. Even though these higher relative humidities are often under control, they have contributed largely to the problem of condensation and frost formation within walls.

In recent years the loss of heat from a building due to the leakage of air around windows and doors has become more fully appreciated and various types of weatherstrips have been extensively used to reduce this exchange of air between the inside and outside of the building. The absolute humidity of the outside air is usually much lower than that of the inside air and the natural result has been to build up the moisture content of the inside air by the normal processes within the building even though artificial humidification is not used.

Increasing the relative humidity of the air in a building or adding weatherstrips to a building will increase the probability of condensation troubles, but the addition of insulation to the walls may either increase or decrease this probability, depending upon the type of construction and the part of the wall under consideration. Since insulation resists the passage of heat, that portion of the wall on the warm side of the insulation will become warmer and that

[1] Condensation in Walls and Attics, by L. V. Teesdale (*U. S. Forest Products Bulletin*).

portion on the cold side will become colder after the insulation is applied. If the inside wall surfaces are being considered the conditions are improved. If, however, consideration is being given to that portion of the wall on the cold side of the insulation, conditions may be aggravated due to the lower temperatures, especially if the moisture can penetrate to this part of the wall.

In many instances there is no logical argument against the use of higher relative humidities for better conditions of comfort and health, nor against the addition of weatherstrips and other means of reducing the air leakage between the inside and outside of a building. Much less could any argument be applied against the addition of insulation. Each of these are essential factors for a well constructed building and a comfortable place in which to live. Each factor may be considered as contributing to the condensation problem, but the elimination of any one would not be a positive cure. The problem must be solved by using a type of construction which will permit the use of all three factors and prevent condensation under all reasonable conditions.

Vapor must be considered as a material by itself independent of the air with which it is mixed. It has the same density, pressure, and dew-point temperature regardless of whether air is present or not. It moves from one place to another without the necessity of moving the air. It passes through some materials very readily while others offer a high resistance to its passage. The amount passing through any material is considered proportional to the drop in vapor pressure along the path of flow through that material, although this law should have further investigation. The following are some of the factors which must be considered in making an analysis of a particular problem: first, the air conditions to be maintained within a building with particular reference to its dry-bulb and dew-point temperatures; second, the outside weather conditions to be expected with reference to wind velocity, minimum temperatures, and duration of low temperature periods; third, the conductivity of the wall under consideration, with particular reference to the probable temperatures of its inner surface and interior sections; and fourth, the vapor proofness of the materials from which the wall is constructed, with particular reference to its inner and outer surface linings.

It is usually not difficult to estimate reasonable inside air temperatures and humidities which are to be carried in the building, but if these requirements are not adhered to in practice they may be directly responsible for condensation difficulties. Since outside weather conditions cannot be controlled an intelligent study of Weather Bureau records should be made to find what extreme conditions may be expected for any particular location. The thermal conductivities of most materials and combinations of materials are sufficiently well understood to make calculations for the wall temperatures to be expected under maximum design conditions. The vapor transmitting properties of various materials under different conditions have not as yet been well defined and this presents one of the most difficult steps in designing a structure which will be free from condensation troubles. It is easy to determine the dew-point temperature of the air which is to be maintained within a building and to calculate the minimum outside weather conditions which may be met without lowering the inside surface temperature below the dew-point temperature of the air, but it is not easy to calculate the dew-point temperature which may be expected for the vapor in the interior parts of the wall. Vapor travels from a point of high vapor pressure to a point of lower vapor pressure, which in the case

of ordinary walls in cold climates is from the inside surface to the outside surface of the wall. If the inner surface has a low resistance to the passage of vapor and the outer surface has a high resistance, then in extreme cases the interior part of the wall may have vapor pressures equal to the air inside of the building. If, on the other hand, the inside surface has a high vapor resistance and the outside a low vapor resistance, then the vapor in the interior parts of the wall may have a dew-point temperature more nearly equal to that of the outside air. Since condensation of moisture can only take place when the wall temperatures are below the dew-point temperature of the vapors present, the logical solution is to select materials for the inside sections of a wall which have a high vapor resistance, and for the outside sections materials which have

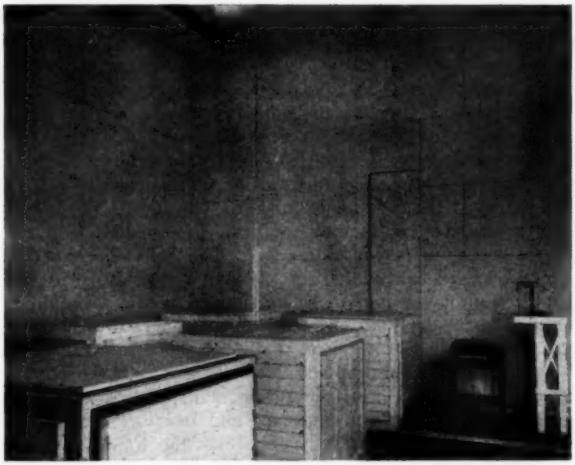

FIG. 1. VIEW OF COLD ROOM SHOWING TEST HOUSE IN PLACE AND WEIGHING BALANCES

a low vapor resistance. In so far as possible the vapor should be stopped on the high temperature side of the wall and not on the low temperature side. In most cases the vapor density within any section of a wall will be governed by the difference in vapor pressure on the two sides of the wall and the vapor resistance of the materials on each side of the section under consideration. By the proper selection and application of materials in a wall the rate of vapor travel through the various parts of the wall may be controlled, and therefore the vapor pressure within the wall may be regulated and maintained at a safe level for any temperature conditions which may be expected within the wall.

Recognizing that there are condensation problems with many types of building construction in combination with various air temperatures and relative humidities as found in modern types of construction, an extensive research program has been initiated to investigate the several factors. The aim is first, to determine the extent of the condensation problem under normal and extreme

conditions; second, to obtain a measure of the relative value of the various methods of correcting these difficulties; and third, to find the limiting conditions under which various types of building construction may be used without excessive condensation troubles and still retain the benefits of higher humidities and good building construction.

It was decided to approach the problem in a practical way and to make tests which would apply to full scale building construction under temperature and humidity conditions which might be expected in the severe climates. To accomplish this a test room was constructed 30 ft square and 25 ft high in which it was possible to control the air temperatures to a minimum of 30 F below zero. Small size test houses were constructed and placed within this room with provisions for maintaining the air temperature and humidities within the

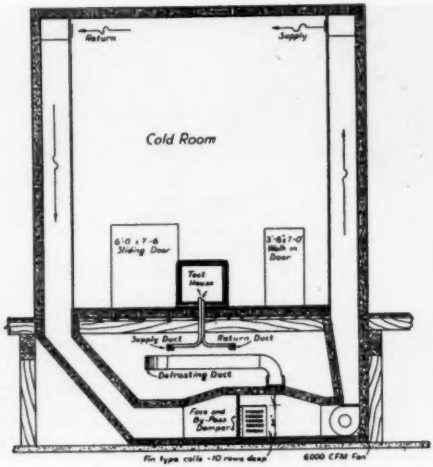

FIG. 2. SECTIONAL VIEW THROUGH COLD ROOM AND COOLING UNIT

houses at any desired level. The houses were so constructed that they might be taken apart and examined from time to time throughout a test period to determine the physical conditions of the interior of the walls and to make comparisons between various types of structures. After fundamental data are obtained covering a sufficient variety of typical construction, larger scale models are to be constructed to prove out on them the value of the more promising designs. The test apparatus has been completed and tests on the small size models have been in progress for approximately two months; the results of these tests will be presented in this discussion.

TEST APPARATUS

The test room is shown in the photograph of Fig. 1 and the line drawing of Fig. 2. The set-up for tests with six small houses in place is shown in

Fig. 1. A vertical cross-sectional view drawn through the center of the room looking toward the front is illustrated in Fig. 2. The room is 30 ft square and 25 ft high and is insulated throughout with 6 in. of rock wool. It is constructed with 2 x 6 studs lined on the inside surface with ½ in. insulating board and no interior finish. The outside surface over the studs is thoroughly sealed with a vapor resisting paper cemented at the joints and finished on the outside with ½ in. insulating board. The 6-in. floor joists are sealed on top and bottom with the vapor proof paper and the space between is filled with rock wool. The object in constructing the room was to prevent the leakage of any outside moist air into the insulation, and to allow a slight amount of breathing through the cold surface. Thus far this type of construction has proved satisfactory and it has been possible to hold the tem-

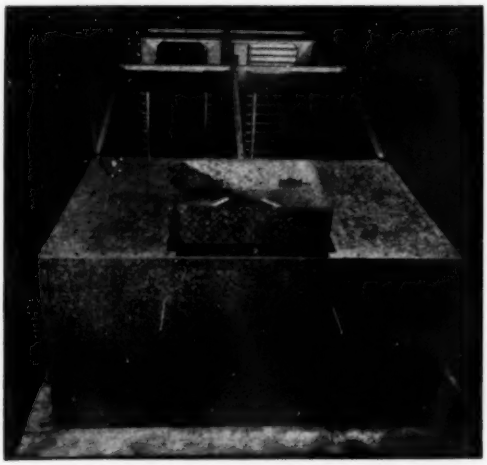

FIG. 3. VIEW OF COOLING UNIT WHILE UNDER CONSTRUCTION

perature in the room to 25 F below zero. Two doors are provided, one of them 6 ft x 7 ft 6 in. used to move large pieces of equipment into and out of the room, and the other a small service door 3 ft 6 in. x 7 ft. The room is provided with nine 10-in. square openings in the floor through which conditioned air may be supplied to each test unit.

The room is cooled by a 25 ton ammonia compressor with direct expansion fin type cooling coils. The cooling unit is placed in the basement under the test room and air is circulated through the cooling unit up through a vertical duct in the right hand corner of the room, Fig. 2, and distributed from a horizontal duct at the ceiling. Air is collected by a similar duct at the ceiling on the opposite side of the room and returned to the cooling unit. A sectional view of the cooling unit is shown below the test room in Fig. 2, a top view from the air discharge end during the period of construction is shown in Fig. 3, and a completed view is shown in the foreground of the photograph of Fig. 4.

Air is circulated through the cooling unit by a 6000 cfm fan located on the discharge side of the unit. The air is drawn through the cooling unit in two parallel streams passing through identically constructed cooling compartments either of which may be completely closed off and connected with recirculated room air for defrosting purposes. A shut-off door is provided at each end of each compartment, and one compartment may be operated independently of the other. Each air passage is provided with a face and by-pass damper at the entrance to a direct expansion ammonia cooling coil. The cooling coil is 10 rows deep, 18 in. wide by 34.5 in. high with a by-pass damper over the top of coils 18 in. wide and 12 in. high. The complete unit is enclosed in a galvanized iron case covered on the outside with 6 in. of rock wool and ½ in. insulating board.

A defrosting fan delivers air at room temperature through either of two ducts shown at the top of the unit in Fig. 4. One of these ducts is connected

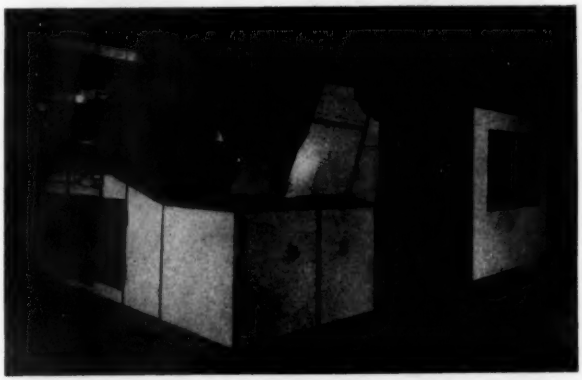

FIG. 4. VIEW OF COMPLETED COOLING UNIT AND ONE END OF CONTROL ROOM

to each cooling compartment and dampers are so arranged that when this cooling compartment is shut off from the main circulating fan the room air may be circulated through the cooling coils and discharged out into the room for defrosting purposes. In practice it has been found necessary to defrost once in two days and a period of 15 min is sufficient for defrosting each side of the equipment.

Conditioned air is supplied to each of the test units in the large cold room through openings in the floor, which are directly under each test unit, from an air conditioning plant located in the basement and shown in the photograph, Fig. 5, and the line drawing, Fig. 6. The conditioning plant controls the humidity and circulates the air through branch ducts distributing it through the floor openings into the small test houses. The air from the test houses is collected and returned to the air conditioning unit for reconditioning. The temperature of the air is controlled independently for each test house by electric heating elements placed in the supply duct and controlled by a thermostat placed in the return air line for the particular unit. The electrical heaters

are arranged so that a part may be connected to a constant supply and a part on a thermostatic control line. Thus the load may be controlled with very little fluctuation in the supply line.

All of the control instruments and thermocouples are brought to a control panel shown in the photograph of Fig. 7 and located in the instrument room which is shown at the right on Fig. 4. At the top of the control panel there are nine switches and pilot lights for the heating elements of the nine test sections located in the cold room. At the right central section is shown the switch and pilot light of the humidity control for the air conditioning unit, and at the central and left central section of the panel are shown the controls for the motor which operates the face and by-pass dampers in the cooling unit to control the temperature in the cold room. At the lower section of

FIG. 5. AIR CONDITIONING UNIT FOR TEST HOUSES

the board there are switches for 60 thermocouples which are wired directly to corresponding panels on the inside walls of the large cold room. Thus thermocouples may be connected directly from the test houses to the panels within the cold room and the readings taken at the control board without the necessity of running wires from the cold room to the control board for each set-up.

Fig. 8 shows at the left a view of the 25 ton ammonia compressor used for the cooling units, at the central sections a view of the control instrument room, and at the right a view of the defrosting fan and discharge ducts. These are located in the basement underneath the large test room.

TEST HOUSES

The test houses were all of frame construction and 4 ft x 4 ft x 4 ft inside dimensions. The main frame for supporting the walls and ceiling of each

house was built up with 4 in. x 4 in. corner posts attached rigidly to the four corners of the floor and connected at the top with a 4 in. x 4 in. frame. The walls were constructed in sections so that they could be inspected for frost formation or other changes in conditions during test periods. Fig. 9 shows a photographic view of two of the frames located in position for test in the cold room with one of the walls applied. Fig. 10 is an oblique sectional drawing of a complete house showing in detail the methods of applying walls to the main frame, and of sealing the interior and exterior joints. Fig. 11 shows an oblique sectional drawing of one of the walls with the sheathing and siding removed as provided for in the method of construction.

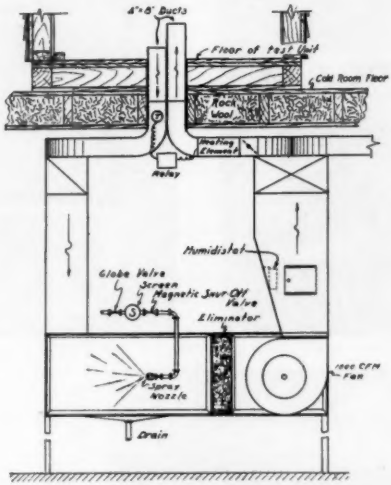

FIG. 6. SECTIONAL VIEW OF AIR CONDITIONING UNIT WITH DISTRIBUTION DUCTS TO TEST HOUSES IN COLD ROOM

The walls were constructed of 2 in. x 4 in. studs spaced 16 in. on center with a 2 in. x 4 in. plate at top and bottom. In all cases the interior finish was made up of metal lath and three coats of plaster; a scratch, brown and finish coat, making a total of ¾ in. in thickness. The outside sections of the walls were constructed with 25/32 in. pine sheathing, building paper, and 6 in. redwood siding. For walls Nos. 1 to 16 inclusive, 6 in. pine butt sheathing was used, and for all other walls 8 in. pine shiplap was used. The outside construction was nailed to 1 in. square nailing strips which were fitted into corresponding grooves in the outside surface of the studs. The assembled outside unit was securely applied to the studs using wood screws instead of nails so that it could be removed for inspection at various periods of the test. The assembled wall was provided on the outside with a 2 in. x 2 in. x ⅛ in. angle iron frame as shown in Fig. 11. When the assembled wall was applied to the test house the angle iron frame was securely fastened and sealed to the 4 in. x 4 in. framework.

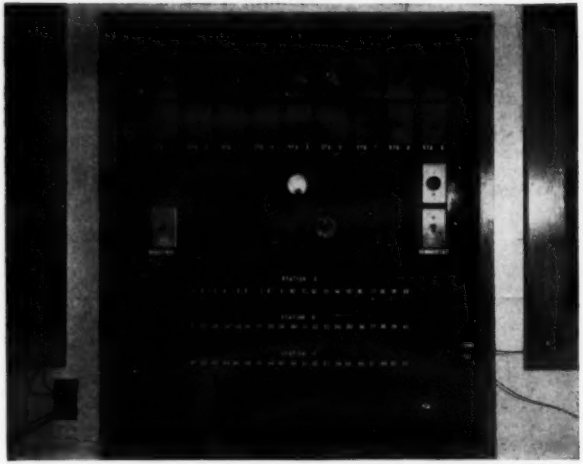

FIG. 7 (*above*). CONTROL AND THERMOCOUPLE PANEL FOR TEST EQUIPMENT

FIG. 8 (*upper right*). AMMONIA COMPRESSOR, CONTROL ROOM AND DEFROSTING FAN FOR COOLING EQUIPMENT

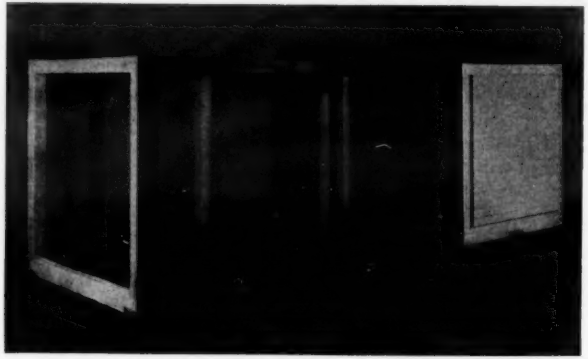

FIG. 9 (*right*). FRAMES FOR TEST HOUSES SET IN COLD ROOM WITH ONE WALL APPLIED

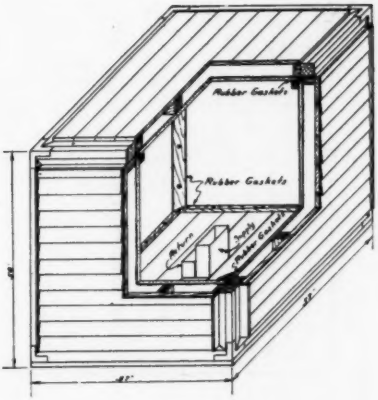

FIG. 10. SECTIONAL VIEW OF TEST HOUSES SHOWING DETAILS OF CONSTRUCTION

The ceilings were constructed with 2 in. x 4 in. joists spaced 16 in. on center with metal lath and 3 coats of plaster on the inner surface, and with building paper and 6 in. pine butt sheathing on the outer surface. The outer surface finish was nailed to 1 in. square nailing strips and fastened to the studs with wood screws in the same manner as for the side wall construction. A 2 in. x 2 in. x ⅛ in. angle iron frame was applied around the outside of the finished ceiling. The floors of the test unit were built with 2 in. x 4 in. joists spaced 16 in. on center with building paper on top and bottom and covered on the lower side with 6 in. butt sheathing, and on the upper side with rough flooring and a finished fir floor. The floors were made integral

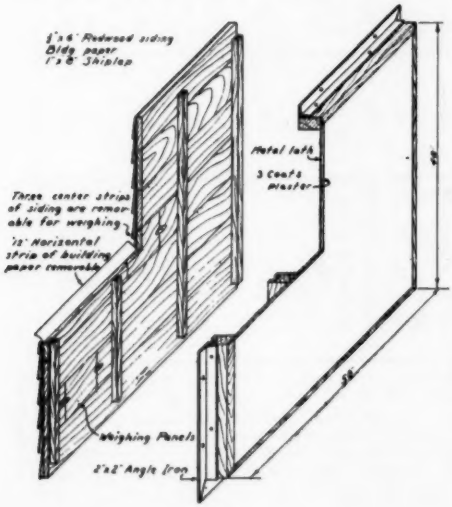

FIG. 11. TYPE OF WALL CONSTRUCTION USED TO FACILITATE INSPECTION

TABLE 1—DETAILS OF WALL CONSTRUCTION[a]

WALL NO.	TEST NO.	VAPOR BARRIER		INSULATION BETWEEN STUDS	ALUMINUM PANEL ON SHEATHING BETWEEN STUDS	NOTES
		Inside Surface of Plaster	Between Metal Lath and Studs			
1	1	No	No	No	No	
1-A	2, 3	2 coats white flat paint	No	3⅝ in. rock wool	[b] Yes	
2	1, 2, 3	No	No	No	Tests 2, 3 only	8 thermocouples in wall
3	1	No	No	No	No	
3-A	2, 3	2 coats aluminum paint	No	3⅝ in. rock wool	[b] Tests 2, 3 only	
4	1	No	No	No	No	
4-A	2, 3	1 coat size; 2-coats white flat paint	No	3⅝ in. rock wool	[b] Yes	
5	1	No	No	3⅝ in. rock wool	No	2-¼ in. holes drilled in top and bottom of outer stud section during test
5-A	2	No	No	3⅝ in. rock wool	Yes	2-¼ in. holes in top and bottom of each stud section
5-B	3	No	No	3⅝ in. rock wool	Yes	2-1 in. holes in top and bottom of each stud section
6	1, 2, 3	No	No	3⅝ in. rock wool	Tests 2, 3 only	8 thermocouples in wall
7	1	No	No	3⅝ in. rock wool	No	2-⅛ in. holes drilled in top and bottom of outer stud section during test
7-A	2, 3	2 coats size	No	3⅝ in. rock wool	[b] Yes	
8	1	No	No	3⅝ in. rock wool	No	

a, b See p. 107

TABLE 1 (*Continued*)—DETAILS OF WALL CONSTRUCTION[a]

8–A	2	1 coat size; 2 coats aluminum paint	No	3⅝ in. rock wool	[b] Yes	
9	1, 2, 3	No	A	3⅝ in. rock wool	Tests 2, 3 only	8 thermocouples in wall, paper lapped at joints
10	1, 2, 3	No	A	3⅝ in. rock wool	Test 3 only	Paper lapped at joints
11	1, 2, 3	No	B	3⅝ in. rock wool	[b] Tests 2, 3, only	8 thermocouples in wall. Paper lapped at joints
12	1, 2, 3	No	B	3⅝ in. rock wool	Test 3 only	Paper lapped at joints
13	1, 2, 3	No	A	3⅝ in. rock wool	[b] Tests 2, 3 only	8 thermocouples in wall. Paper sealed at joints, with asphalt
14	1, 2, 3	No	A	3⅝ in. rock wool	Test 3 only	Paper sealed at joints with asphalt
15	1, 2, 3	No	B	3⅝ in. rock wool	Test 3 only	Paper sealed at joints with asphalt
16	1, 2, 3	No	B	3⅝ in. rock wool	Tests 2, 3 only	Paper sealed at joints with asphalt
17	3	No	Bldg. Paper	3⅝ in. rock wool	Yes	
18	3	No	Bldg. Paper	3⅝ in. rock wool	Yes	
19	3	No	No	3⅝ in. rock wool	Yes	
20	3	No	No	3⅝ in. rock wool	Yes	No bldg. paper between sheathing and siding
21	3	No	No	3⅝ in. rock wool	Yes	1–2 in. hole at top of each stud section
24	3	1 coat asphalt	No	3⅝ in. rock wool	Yes	

[a] All walls were built with 2 in. x 4 in. studs spaced 16 in. O. C., metal lath and three coats of gypsum plaster on inside surface, 25/32 in. pine sheathing, 50 lb building paper, 6 in. redwood siding with three coats white lead paint on outside surface. Six inch pine butt sheathing was used in walls No. 1 to 16, inclusive, and 8 in. pine shiplap sheathing was used in all other walls. Vapor barriers were added to the inside section, insulation was added between the studs, and paper was removed from or holes were provided in outside section as indicated for the various walls. In part of the walls aluminum panels were placed against the inside surface of the sheathing between the studs as a means of collecting and weighing condensate. Thermocouples were placed between the aluminum panel and sheathing.

[b] Aluminum panels changed from outer stud section to center stud section between tests 2 and 3.

A = 30, 30, 30 duplex paper, two sheets of 30 lb paper cemented together with 30 lb asphalt.

B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

TABLE 2—DETAILS OF CEILING CONSTRUCTION[a]

CEILING No.	TEST No.	VAPOR BARRIER		INSULATION BETWEEN JOISTS	ALUMINUM PANEL ON SHEATHING BETWEEN JOISTS	NOTES
		Inside Surface of Plaster	Between Metal Lath and Studs			
C-1	1, 2	No	No	No	No	
C-1-A	3	No	No	3⅝ in. expanded aluminum-magnesium silicate	Yes	
C-2	1, 2, 3	No	No	3⅝ in. rock wool	[b] Tests 2, 3 only	
C-3	1, 2, 3	No	B	3⅝ in. rock wool	[b] Tests 2, 3 only	Paper lapped at joints. 5 thermocouples in ceiling
C-4	1	No	B	3⅝ in. rock wool	No	Paper sealed at joints with asphalt
C-4	2, 3	No	B	3⅝ in. rock wool	Yes	Paper sealed at joints with asphalt
C-5	3	No	No	3⅝ in. redwood bark fiber	Yes	
C-6	3	No	No	3⅝ in. sawdust	Yes	

[a] All ceilings were built with 2 in. x 4 in. joists spaced 16 in. O. C., metal lath and three coats of gypsum plaster on inside surface, 50 lb building paper between joists and sheathing, no siding. Sheathing painted with three coats of white lead paint on outside surface. Six inch pine butt sheathing was used in ceilings 1 to 4 inclusive, and 8 in. pine shiplap sheathing was used in all other ceilings. Vapor barriers were added to the inside section and insulation was added between the joists as indicated. Aluminum panels supplied as described in Table 1.

[b] Aluminum panels changed from outer joist section to center joist section between tests 2 and 3.

B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

with the 4 in. x 4 in. framework and not removed or opened for inspection during the test.

The first and second coats of plaster were made of beaver fibered plaster and the finished coat was ivory hydrated finishing lime. In all cases the plaster was allowed to dry at least 14 days before making tests. The paint used for the outside surface of the walls was made up of 70 per cent pigment and 30 per cent vehicle by weight. The pigment was a pure white lead paste and the vehicle was 90 per cent linseed oil and 10 per cent dryer.

As indicated in Table 1 showing wall construction, some of the walls were insulated and some were not. Vapor proof barriers were added to the inside surface of part of the walls in the form of a surface paint or in the form of a vapor proof paper placed between the metal lath and studs. The outside sections of some of the walls were changed by omitting the building paper and by drilling ventilating holes through the sheathing.

When a test unit was assembled by applying the walls and ceiling to the main frame, the joints between the angle iron frame around each wall and the main frame of the test unit were sealed with putty, and the joints along the interior corners were sealed with sealing strips and rubber gaskets to prevent the infiltration of air. The test house was set in the cold room and thoroughly sealed to the floor to prevent air leakage under the floor of the test house. The conditioned air was circulated through the test house by the ducts as shown in Fig. 10.

For each wall three sections of the sheathing, one board in width and 8 in. long, were cut out during the construction, closely fitted into place and applied with turn buttons. The loose sections were removed and weighed at periods during the tests to measure the amount of moisture and frost accumulated on the sheathing.

After the first series of tests there was some question as to whether or not some of the vapor which passed through the inner section of the wall was also leaving the walls through the exterior finish. In order to reduce this possibility sheets of aluminum were placed between the studs and against the sheathing, one such sheet being used for each wall. Thermocouples were placed throughout part of the walls and in each case a thermocouple was placed between the aluminum sheet and the wood sheathing in close contact with the aluminum sheet. The temperature indicated by this thermocouple was taken as the temperature of the aluminum sheet.

In a few of the insulated walls a central section of the siding and building paper was so applied that it could be removed for inspection during a test.

TEST PROCEDURE

In making a test the test houses were completely assembled, the cooling equipment started, and the room temperature brought down to test requirements. Test conditions were then maintained for a reasonable time to establish uniform temperature and humidity gradients through the walls. After this preliminary period the walls were inspected to determine their condition without allowing any unnecessary changes in temperature or moisture conditions surrounding them. After these records were made the walls were again closed and the test continued for a period of from 5 to 19 days, after which the walls were again taken apart and inspected to get the rate of frost

TABLE 3—TEST NO. 1—DATA AND RESULTS

WALL NO.	PHOTOGRAPH TAKEN FIG. NO.	COLD ROOM F	TEST HOUSE		WALL TEMPERATURE F								PER CENT CHANGE IN WEIGHT OF INSULATION	MOISTURE GAIN GRAMS PER SQ FT PER 24 HOURS		
					Plaster		Air Space			Sheathing		Outside Surface of Siding				
			D. B. Temp. F	Per Cent R. H.	Inside Surface	Outside Surface	Top	Center	Bottom	Inside Surface	Outside Surface			Sheathing	Bldg. Paper	Siding
Test Period No. 1—100 Hours																
1		−19.2	70.3	41	...	...	...	...	...	...	...	...	...	...	...	...
2		−19.2	70.3	41	62.0	55.6	50.8	46.2	33.1	38.5	16.2	−2.0	...	...	...	...
3		−19.2	70.3	41	...	...	...	...	...	...	...	...	...	...	...	...
4	12	−19.2	70.3	41	...	...	...	...	...	...	...	...	...	1.29	...	...
C-1		−19.2	70.3	41	...	...	...	...	...	...	...	...	...	−0.11	...	...
5		−19.2	70.5	41	...	...	...	...	...	...	...	...	...	...	...	...
6		−19.2	70.5	41	68.1	65.7	27.3	26.8	22.8	1.6	−5.9	−11.7	...	...	...	...
7		−19.2	70.5	41	...	...	...	...	...	...	...	...	...	...	...	...
8		−19.2	70.5	41	...	...	...	...	...	...	...	...	−0.99	4.07	...	...
C-2		−19.2	70.5	41	...	...	...	...	...	...	...	...	...	−0.23	...	...
9		−19.2	70.7	40	65.1	62.3	25.0	20.8	21.6	−3.6	−9.9	−16.0	...	...	...	...
10		−19.2	70.7	40	...	...	...	...	...	...	...	...	...	1.64	...	...
11		−19.2	70.7	40	66.3	61.6	24.6	22.6	15.4	−3.6	−9.9	−17.2	...	...	...	...
12		−19.2	70.7	40	...	...	...	...	...	...	...	...	...	0.35	...	...
C-3		−19.2	70.7	40	71.4	67.2	...	24.7	...	−8.1	...	−13.4	...	...	...	...
13		−19.2	70.3	40	67.9	64.0	23.2	19.6	9.9	−1.6	−7.6	−15.9	...	...	...	...
14		−19.2	70.3	40	...	...	...	...	...	...	...	...	...	0.58	...	...
15		−19.2	70.3	40	...	...	...	...	...	...	...	...	...	...	...	...
16		−19.2	70.3	40	...	...	...	...	...	...	...	...	...	0.23	...	...
C-4		−19.2	70.3	40	...	...	...	...	...	...	...	...	...	1.29	...	...

TABLE 3 (*Continued*)—TEST NO. 1—DATA AND RESULTS

WALL NO.	PHOTOGRAPH TAKEN FIG. NO.	COLD ROOM F	TEST HOUSE		WALL TEMPERATURE F								PER CENT CHANGE IN WEIGHT OF INSULATION	MOISTURE GAIN GRAMS PER SQ FT PER 24 HOURS		
			D. B. Temp. F	Per Cent R. H.	Plaster		Air Space			Sheathing		Outside Surface of Siding		Sheathing	Bldg. Paper	Siding
					Inside Surface	Outside Surface	Top	Center	Bottom	Inside Surface	Outside Surface					
Test Period No. 2—360 Hours																
1		−15.2	70.0	44	...	...	...	...	...	...	...	...	...	...	...	...
2		−15.2	70.0	44	62.0	55.7	50.9	47.1	34.5	38.9	19.2	2.7	...	...	...	...
3		−15.2	70.0	44	...	...	...	...	...	...	...	...	...	0.92	...	...
4[a]		−15.2	70.0	44	...	...	...	...	...	...	...	...	...	0.89	0.125	0.13
C-1[a]	25	−15.2	70.0	44	...	...	...	...	...	...	...	...	...	0.44	...	...
5	13	−15.2	70.0	44	...	...	...	...	...	...	...	...	...	...	...	...
6		−15.2	70.0	44	67.9	65.2	28.6	28.8	26.6	5.5	−1.0	−6.3	...	...	...	...
7		−15.2	70.0	44	...	...	...	...	...	...	...	...	...	...	...	...
8		−15.2	70.0	44	...	...	...	...	...	...	...	...	−1.81	2.74	0.010	0.015
C-2[a]	26	−15.2	70.0	44	...	...	...	...	...	...	...	...	...	0.23	...	...
9		−15.2	70.1	43	65.4	62.5	25.5	23.3	23.3	−0.1	−6.1	−11.7	...	...	...	...
10[a]	14	−15.2	70.1	43	...	...	...	...	...	...	...	...	−0.53	1.27	0.010	0.010
11		−15.2	70.1	43	66.4	61.9	25.8	25.2	19.3	0.6	−5.3	−11.9	...	...	...	...
12[a]	15	−15.2	70.1	43	...	...	...	...	...	...	...	...	−0.67	0.96	0.005	0.016
C-3		−15.2	70.1	43	71.3	67.2	...	26.8	...	−4.0	...	−8.8	...	...	...	...
13		−15.2	70.3	45	67.7	63.9	24.1	21.8	13.4	2.2	−3.5	−11.1	...	...	...	...
14		−15.2	70.3	45	...	...	...	...	...	...	...	...	−0.39	0.53	0.010	0.016
15		−15.2	70.3	45	...	...	...	...	...	...	...	...	...	...	...	...
16[a]	16	−15.2	70.3	45	...	...	...	...	...	...	...	...	−0.19	0.39	0.005	0.021
C-4[a]	27	−15.2	70.3	45	...	...	...	...	...	...	...	...	...	0.44	...	...

[a] Photographs taken at end of test period No. 2.

TABLE 3 (*Continued*)—TEST NO. 1—DATA AND RESULTS

WALL NO.	PHOTOGRAPH TAKEN FIG. NO.	COLD ROOM F	TEST HOUSE		WALL TEMPERATURE F								PER CENT CHANGE IN WEIGHT OF INSULATION	MOISTURE GAIN GRAMS PER SQ FT PER 24 HOURS		
			D. B. Temp. F	Per Cent R. H.	Plaster		Air Space			Sheathing		Outside Surface of Siding		Sheathing	Bldg. Paper	Siding
					Inside Surface	Outside Surface	Top	Center	Bottom	Inside Surface	Outside Surface					
Test Period No. 3—120 Hours																
1		−9.9	70	46	...	...	...	...	...	...	...	...	...	...	...	...
2		−9.9	70	46	62.0	56.3	51.7	48.1	36.1	40.9	21.6	5.2	...	...	...	...
3		−9.9	70	46	...	...	...	...	...	...	...	...	...	0.85	0.145	0.09
4[a]		−9.9	70	46	...	...	...	...	...	...	...	...	...	0.83	0.25	0.14
C-1[a]		−9.9	70	46	...	...	...	...	...	...	...	...	...	0.38	...	...
5[a]		−9.9	70.4	45	...	...	...	...	...	...	...	...	−6.24	4.25	0.012	0.025
6		−9.9	70.4	45	68.4	65.8	31.6	30.5	30.0	8.3	2.5	−2.5	...	...	...	...
7		−9.9	70.4	45	...	...	...	...	...	...	...	...	...	...	...	...
8		−9.9	70.4	45	...	...	...	...	...	...	...	...	−2.44	2.72	0.017	0.021
C-2[a]		−9.9	70.4	45	...	...	...	...	...	...	...	...	...	0.24	...	...
9		−9.9	69.9	47	65.3	62.6	27.4	25.4	25.1	3.6	−1.9	−7.0	−0.12	0.64	0.008	...
10[a]		−9.9	69.9	47	...	...	...	...	...	...	...	...	−1.16	1.03	0.008	0.021
11		−9.9	69.9	47	66.3	62.1	26.4	27.8	22.4	4.3	−1.2	−7.2	−0.03	0.34	0.004	0.012
12[a]		−9.9	69.9	47	...	...	...	...	...	...	...	...	−0.80	0.63	0.004	0.012
C-3		−9.9	69.9	47	71.4	67.3	...	29.2	...	0.5	...	−4.1	...	...	...	...
13		−9.9	70.2	45	68.0	64.4	26.1	24.4	17.0	5.9	0.5	−6.5	−0.83	0.34	0.004	0.025
14		−9.9	70.2	45	...	...	...	...	...	...	...	...	−0.71	0.56	0.008	0.012
15		−9.9	70.2	45	...	...	...	...	...	...	...	...	−0.86	0.30	0.004	0.021
16[a]		−9.9	70.2	45	...	...	...	...	...	...	...	...	−0.67	0.42	0.004	0.008
C-4[a]		−9.9	70.2	45	...	...	...	...	...	...	...	...	...	0.36	...	...

[a] Photographs taken at end of test period No. 2.

formation or other changes under constant temperature and humidity conditions. At the end of the test the walls were completely taken apart and all observations made including photographs of typical conditions. The following tests have been run and results given in Tables 3, 4 and 5.

Test 1, first period,	Oct.	6 to	Oct. 11	100 hours
Test 1, second period,	Oct.	11 to	Oct. 27	360 hours
Test 1, third period,	Oct.	27 to	Nov. 2	120 hours
Test 2, first period,	Nov.	6 to	Nov. 12	136 hours
Test 2, second period,	Nov.	12 to	Nov. 17	114 hours
Test 3, first period,	Nov.	19 to	Nov. 22	80 hours
Test 3, second period,	Nov.	22 to	Nov. 27	120 hours

As previously stated one of the objects of these tests was to demonstrate some of the causes of condensation in walls, the extent to which it is likely to occur under severe conditions, and some practical methods of overcoming the difficulty without foregoing the comfort and benefits of either air conditioning or good building construction. The tests have been made under conditions which were severe enough and for a sufficient length of time to show the nature of the condensation problems and to demonstrate that there are practical methods of meeting these problems. The methods of measuring the extent of the vapor travel through various types of wall construction have been relative and not reduced to an absolute basis. In the first part of the tests the rate of the vapor travel was measured by observation and weight of the frost formed on the inside surface of the sheathing for the different walls. For the last test it was measured by placing an aluminum condensation panel between the studs in contact with the inside surface of the wood sheathing. The test results, however, showed a reasonable check between the amount of moisture collected on the aluminum panel and that collected on the sheathing for similar sections of a wall of the same construction and under the same test conditions. The greatest error in the collection of moisture by this method appeared to be in the uninsulated walls. In these cases the inner surfaces of the sheathing were not far from freezing temperatures and the moisture was not retained on the surface of the sheathing as frost but seemed to penetrate and pass through. The difference between the vapor pressure of the air on the two sides of sheathing was greater for the uninsulated wall than it was for the insulated walls. Thus the vapor which did travel through the inner finish had a greater tendency to travel through the outer finish and not be collected. In these tests the temperature of the inner surface of the outside sheathing has been taken as a measure of the dew-point temperature of the air within the space between the studs. This is sufficiently accurate for comparative purposes, but from an investigation which is now under way it appears that the actual dew-point temperature of the air in the space between the studs is a considerable amount higher than the surface temperatures of the sheathing and that for absolute results the vapor pressure must be measured. While there is a great deal more test work necessary to determine absolute values for various types of construction, and to find the most practical applications for different outside weather and inside air conditions, the work has been carried far enough to show that there are practical solutions of the condensation problem and that when conditions are thoroughly understood it will be only a matter of selecting material and types of construction

TABLE 4—TEST NO. 2—DATA AND RESULTS

WALL NO.	COLD ROOM TEMP. F	TEST HOUSE		WALL TEMPERATURES F								PER CENT CHANGE IN WEIGHT OF INSULATION 250 HOURS	MOISTURE GAIN GRAMS PER SQ FT PER 24 HOURS		
				Plaster		Air Space			Sheathing				Sheathing 250 HOURS	Aluminum Panels 250 Hours	Aluminum Panels by Periods
		Dry-Bulb Temp. F	Per Cent R. H.	Inside Surface	Outside Surface	Top	Center	Bottom	Inside Surface	Outside Surface	Outside Surface of Siding				
Test Period No. 1—136 Hours															
1-A	−8.9	69.8	43	...	...	...	...	...	...	...	...	...	...	...	1.99
2	−8.9	69.8	43	62.9	57.0	52.5	48.9	36.6	41.5	21.8	6.8	...	...	...	0.19
3-A	−8.9	69.8	43	...	...	...	...	...	...	...	...	...	...	...	1.42
4-A	−8.9	69.8	43	...	...	...	...	...	...	...	...	...	...	...	1.59
C-1	−8.9	69.8	43	...	...	...	...	...	...	...	...	...	...	...	...
5-A	−8.9	70.1	43	...	...	...	...	...	...	...	...	...	...	...	2.48
6	−8.9	70.1	43	67.8	64.8	32.9	33.6	28.3	8.9	2.8	−2.1	...	...	...	2.83
7-A	−8.9	70.1	43	...	...	...	...	...	...	...	...	...	...	...	1.16
8-A	−8.9	70.1	43	...	...	...	...	...	...	...	...	...	...	...	0.89
C-2	−8.9	70.1	43	...	...	...	...	...	...	...	...	...	...	...	3.0
9	−8.9	70.8	42	65.9	62.9	28.1	28.0	23.0	4.4	−1.4	−6.1	...	...	...	0.43
10	−8.9	70.8	42	...	...	...	...	...	...	...	...	...	...	...	...
11	−8.9	70.8	42	66.9	62.3	24.2	27.9	21.1	5.1	−0.5	−6.6	...	...	...	0.16
12	−8.9	70.8	42	...	...	...	...	...	...	...	...	...	...	...	...
C-3	−8.9	70.8	42	72.3	68.2	...	29.4	...	1.1	−3.5	...	...	...	...	0.17
13	−8.9	70.4	42	68.3	64.5	25.5	23.2	20.0	5.9	0.7	−5.9	...	...	...	0.23
14	−8.9	70.4	42	...	...	...	...	...	...	...	...	...	...	...	...
15	−8.9	70.4	42	...	...	...	...	...	...	...	...	...	...	...	...
16	−8.9	70.4	42	...	...	...	...	...	...	...	...	...	...	...	0.04
C-4	−8.9	70.4	42	...	...	...	...	...	...	...	...	...	...	...	0.19

TABLE 4 (*Continued*)—TEST NO. 2—DATA AND RESULTS

WALL No.	COLD ROOM TEMP. F	TEST HOUSE		WALL TEMPERATURES F								PER CENT CHANGE IN WEIGHT OF INSULATION 250 HOURS	MOISTURE GAIN GRAMS PER SQ FT PER 24 HOURS		
				Plaster		Air Space			Sheathing						
		Dry-Bulb Temp. F	Per Cent R. H.	Inside Surface	Outside Surface	Top	Center	Bottom	Inside Surface	Outside Surface	Outside Surface of Siding		Sheathing 250 Hours	Aluminum Panels 250 Hours	Aluminum Panels by Periods
Test Period No. 2—114 Hours															
1-A	−11.4	69.7	41	...	...	...	...	...	...	...	...	0.0	...	1.64	1.25
2	−11.4	69.7	41	64.2	58.8	55.0	47.8	36.4	30.4	12.9	1.7	...	...	0.53	0.93
3-A	−11.4	69.7	41	...	...	...	...	...	...	...	...	−0.23	0.94	1.47	1.51
4-A	−11.4	69.7	41	...	...	...	...	...	...	...	...	−0.10	2.04	1.37	1.12
C-1	−11.4	69.7	41	...	...	...	...	...	...	...	...	...	0.62	...	...
5-A	−11.4	70.4	40	...	...	...	...	...	...	...	...	0.06	2.98	2.69	3.03
6	−11.4	70.4	40	68.1	64.9	31.9	33.5	28.5	7.9	0.9	−4.4	0.04	...	2.74	2.66
7-A	−11.4	70.4	40	...	...	...	...	...	...	...	...	−0.06	...	1.02	0.86
8-A	−11.4	70.4	40	...	...	...	...	...	...	...	...	0.14	0.78	0.65	0.37
C-2	−11.4	70.4	40	...	...	...	...	...	...	...	...	...	0.16	3.16	3.35
9	−11.4	70.0	40	65.4	62.4	26.6	26.6	22.6	2.1	−3.8	−8.2	0.05	0.32	0.46	0.51
10	−11.4	70.0	40	...	...	...	...	...	...	...	...	...	0.48	...	...
11	−11.4	70.0	40	66.4	62.0	24.6	29.5	20.7	1.7	−3.8	−9.2	−0.05	0.16	0.23	0.33
12	−11.4	70.0	40	...	...	...	...	...	...	...	...	...	0.32	...	...
C-3	−11.4	70.0	40	71.9	67.4	...	27.8	...	−1.1	−5.8	...	...	0.00	0.17	0.18
13	−11.4	70.4	40	68.3	64.7	25.1	25.6	20.5	3.6	−2.5	−8.6	−0.18	0.32	0.17	0.09
14	−11.4	70.4	40	...	...	...	...	...	...	...	...	...	0.16	...	...
15	−11.4	70.4	40	...	...	...	...	...	...	...	...	...	0.32	...	...
16	−11.4	70.4	40	...	...	...	...	...	...	...	...	−0.44	0.16	0.02	0.00
C-4	−11.4	70.4	40	...	...	...	...	...	...	...	...	...	0.16	0.14	0.09

TABLE 5—TEST NO. 3—DATA AND RESULTS

WALL NO.	PHOTOGRAPH TAKEN FIG. NO.	TEST PERIOD NO. 1—80 HOURS					TEST PERIOD NO. 2—120 HOURS				
		Cold Room F	Test House		Inside Surface Temperature of Sheathing F	Moisture Gain on Aluminum Panel, Grams per Sq Ft per 24 Hours	Cold Room F	Test House		Inside Surface Temperature of Sheathing F	Moisture Gain on Aluminum Panel, Grams per Sq Ft per 24 Hours
			Dry-Bulb F	Per Cent R. H.				Dry-Bulb F	Per Cent R. H.		
1-A		−7.4	70.3	40	9.8	1.11	−7.7	70.0	40	9.7	0.31
2[a]	17	−7.4	70.3	40	29.5	0.13	−7.7	70.0	40	28.5	0.46
3-A		−7.4	70.3	40	10.7	0.98	−7.7	70.0	40	10.6	0.23
4-A		−7.4	70.3	40	10.5	1.31	−7.7	70.0	40	10.7	0.33
C-1A		−7.4	70.3	40	9.6	2.56	−7.7	70.0	40	9.2	2.45
5-A		−7.4	70.0	40	3.8	0.0	−7.7	70.4	40	4.3	0.0
6[a]	18	−7.4	70.0	40	10.3	2.26	−7.7	70.4	40	10.2	2.34
7-A[a]	24	−7.4	70.0	40	11.5	0.75	−7.7	70.4	40	11.8	0.22
8-A		−7.4	70.0	40	13.5	0.82	−7.7	70.4	40	13.4	0.22
C-2[a]	28	−7.4	70.0	40	5.9	2.56	−7.7	70.4	40	6.9	2.68
9		−7.4	70.5	39.5	7.3	0.24	−7.7	69.9	40	7.6	0.21
10[a]	20	−7.4	70.5	39.5	9.1	0.20	−7.7	69.9	40	9.2	0.19
11		−7.4	70.5	39.5	6.2	0.00	−7.7	69.9	40	5.8	0.0
12[a]	21	−7.4	70.5	39.5	8.7	0.09	−7.7	69.9	40	9.1	0.06
C-3		−7.4	70.5	39.5	3.8	0.03	−7.7	69.9	40	4.2	0.02
13		−7.4	70.1	40	6.8	0.13	−7.7	70.4	40	7.5	0.11
14[a]	22	−7.4	70.1	40	9.1	0.13	−7.7	70.4	40	9.5	0.12
15		−7.4	70.1	40	8.0	0.0	−7.7	70.4	40	8.4	0.0
16[a]	23	−7.4	70.1	40	8.4	0.0	−7.7	70.4	40	8.8	0.0
C-4[a]	29	−7.4	70.1	40	5.8	0.13	−7.7	70.4	40	6.8	0.15
17		−7.4	70.0	40	7.2	0.26	−7.7	70.0	40	8.1	0.46
18[a]	19	−7.4	70.0	40	8.9	0.42	−7.7	70.0	40	9.3	0.56
19		−7.4	70.0	40	8.9	2.36	−7.7	70.0	40	9.6	2.71
20		−7.4	70.0	40	9.7	2.48	−7.7	70.0	40	10.2	3.02
C-5		−7.4	70.0	40	7.1	4.12	−7.7	70.0	40	7.4	2.96
21		−7.4	70.2	40	11.3	1.77	−7.7	69.9	40	10.7	3.08
24		−7.4	70.2	40	11.0	0.88	−7.7	69.9	40	11.1	0.15
C-6		−7.4	70.2	40	8.7	1.31	−7.7	69.9	40	9.6	2.45

[a] Photographs taken at the end of test.

to insure satisfactory results. The following comparisons taken from the test data are of interest as indicating what may be expected under various combinations of construction and operating conditions.

DIFFERENT TYPES OF CONSTRUCTION COMPARED

Table 1 gives the details of the walls as tested, and Table 2 gives the details of the ceilings. The test data for tests Nos. 1, 2 and 3 are shown in Tables 3, 4 and 5, respectively. As a matter of convenience, Table 6 has been arranged showing the results obtained for the different walls during the second period of the third test. These were all tested under the same conditions and the results showing condensate in grams per square foot per 24 hours were taken as the amount collected on the aluminum sheet placed next to the sheathing in the center stud section. Thus the values are all on the same basis and show a definite comparison between various types of construction. The results for the uninsulated wall No. 2 compared with the insulated walls Nos. 6 and 19 show clearly what may happen when insulation is added to a wall without giving any consideration to the relative vapor resistance of the different sections of that wall. These walls were of a common type of construction with metal lath and plaster on inside surface, and sheathing, building paper, and lap siding on the outside surface. This construction gives a low vapor resistance for the inside surface and a relatively high resistance at the outer surface. Thus the vapor density may build up at the outer surface. In addition to this the inside surface finish of the wall has but very little insulating value and for the uninsulated wall the heat travels through to the outside finish and raises the inside surface of the sheathing to a much higher temperature than will be the case when insulation is used. Thus the possibility of condensation of moisture on the sheathing of the uninsulated wall is reduced, but at the same time the heat losses through the wall are quite high.

The photographs of Figs. 17 and 18 show the conditions which existed after this test. The uninsulated wall, Fig. 17, shows both the wood sheathing and aluminum panel at center section to be free from moisture excepting at the lower part of the wall through which this photograph was taken. The accumulation of frost extended approximately 6 in. from the bottom on the aluminum panel and around the lower outside corner of the wood sheathing. For the insulated wall, Fig. 18, the frost was rather uniformly distributed over the entire aluminum panel and also over the wood sheathing in the exposed section. That part of the wood sheathing shown in the photograph has slightly heavier frost formation than the average for the entire area. The weight of frost collected per square foot of aluminum panel for the uninsulated wall No. 2 was 0.46 grams per 24 hours, and for the insulated wall No. 6 it was 2.34 grams per 24 hours. The results for these two walls show a condition often met in practice and misinterpreted.

The advantage to be gained by adding vapor barriers to the inside surface of a wall is shown by the various photographs and weights of condensate indicated in Table 6 for different types of barriers. A 50 lb standard building paper placed between the metal lath and studs of insulated walls Nos. 17 and 18 showed 0.46 and 0.56 grams of condensate respectively as compared with 2.34 and 2.71 for insulated walls Nos. 6 and 19 respectively, which were of the same construction but with no vapor barrier between the metal lath and studs.

FIG. 12. WALL NO. 4

FIG. 13. WALL NO. 5

FIG. 14. WALL NO. 10

FIG. 15. WALL NO. 12

FIG. 16. WALL NO. 16

FIG. 17. WALL NO. 2

FIG. 18. WALL NO. 6

FIG. 19. WALL NO. 18

FIG. 20. WALL NO. 10

TABLE 6—RELATIVE CONDENSATION FOR DIFFERENT TYPES OF WALL CONSTRUCTION UNDER SIMILAR TEST CONDITIONS

TYPE OF WALL AND INSULATION	SPECIAL VAPOR BARRIER	WALL NO.	PHOTO AFTER TEST FIG. NO.	TEST NO. AND PERIOD NO.	COLD ROOM TEMP. F	SURFACE TEMP. OF SHEATHING F	CONDENSATE GRAMS PER SQ FT PER 24 HOURS
Standard, No insulation	None	2	17	3–2	– 7.7	28.5	0.46
Standard with 3⅝ in. rock wool	None	19	..	3–2	– 7.7	9.6	2.71
Standard with 3⅝ in. rock wool	50 lb. Bldg. Paper,[a] lapped joints	17	..	3–2	– 7.7	8.1	0.46
		18	19	3–2	– 7.7	9.3	0.56
Standard with 3⅝ in. rock wool	A[a] lapped joints	9	..	3–2	– 7.7	7.6	0.21
		10	20	3–2	– 7.7	9.2	0.19
Standard with 3⅝ in. rock wool	B[a] lapped joints	11	..	3–2	– 7.7	5.8	0.0
		12	21	3–2	– 7.7	9.1	0.06
Standard with 3⅝ in. rock wool	A[a] sealed joints	13	..	3–2	– 7.7	7.5	0.11
		14	22	3–2	– 7.7	9.5	0.12
Standard with 3⅝ in. rock wool	B[a] sealed joints	15	..	3–2	– 7.7	8.4	0.0
		16	23	3–2	– 7.7	8.4	0.0
Standard with 3⅝ in. rock wool	2 coats flat white paint	1–A	..	3–2	– 7.7	9.7	0.31
Standard with 3⅝ in. rock wool	2 coats aluminum paint	3–A	..	3–2	– 7.7	10.6	0.23
Standard with 3⅝ in. rock wool	1 coat size, 2 coats flat white paint	4–A	..	3–2	– 7.7	10.7	0.33
Standard with 3⅝ in. rock wool	2 coats size	7–A	24	3–2	– 7.7	11.8	0.22
Standard with 3⅝ in. rock wool	1 coat size, 2 coats aluminum paint	8–A	..	3–2	– 7.7	13.4	0.22
Standard with 3⅝ in. rock wool	1 coat asphalt paint	24	..	3–2	– 7.7	11.1	0.15
Standard with 3⅝ in. rock wool	2 in. hole in sheathing only	21	..	3–2	– 7.7	10.7	3.08
No. Bldg. Paper between sheathing and siding 3⅝ in. rock wool	None	20	..	3–2	– 7.7	9.6	3.06
Standard with 3⅝ in. rock wool	4–1 in. vent holds in each stud section	5–A	..	3–2	– 7.7	4.3	0.0

[a] All vapor barriers placed between metal lath and studs.
A = 30, 30, 30 duplex paper, 2 sheets 30 lb paper cemented together with 30 lb asphalt.
B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

The photograph of Fig. 19 shows the condition of the aluminum panel for wall No. 18 after the test. The sheathing showed no signs of condensation, and that on the aluminum panel was very light as indicated by the weight.

Walls Nos. 9 and 10 and photograph of Fig. 20 show the effect of placing the barrier *A*, known as 30,30,30 duplex paper, between the metal lath and studs. In this case there were 0.21 and 0.19 grams of condensate for 24 hours. No frost appeared on the wood sheathing, and that on the aluminum panel was very light and scattered. The effect of placing an asphalt impregnated and glossy surfaced coated sheathing paper, with joints lapped but not sealed,

XUM

FIG. 21. WALL NO. 12

FIG. 22. WALL NO. 14

FIG. 23. WALL NO. 16

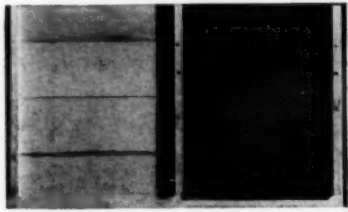

FIG. 24. WALL 7-A

FIG. 25. CEILING NO. C-1

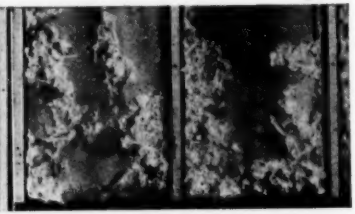

FIG. 26. CEILING NO. C-2

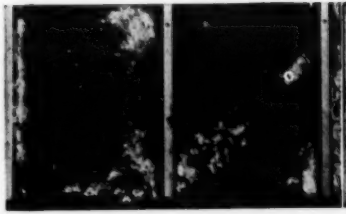

FIG. 27. CEILING NO. C-4

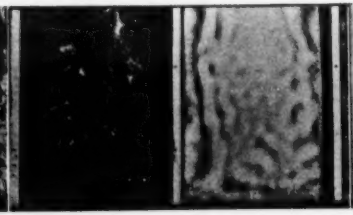

FIG. 28. CEILING NO. C-2

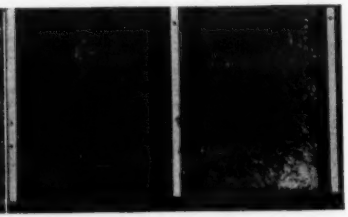

FIG. 29. CEILING NO. C-4

between the metal lath and studs of an insulated wall is shown by the results for walls Nos. 11 and 12, and photograph of Fig. 21. For wall No. 11 the condensate was zero and for wall No. 12, 0.06 grams per 24 hours. The photograph of wall No. 12 shows a very slight amount of condensate collected at the lower part of the aluminum panel but none on the sheathing. A slightly better result was obtained when the two vapor barriers, *A* and *B*, were sealed at the joints, as shown in walls Nos. 13 and 14 and photograph of Fig. 22, for barrier *A*, and walls Nos. 15 and 16, photograph of Fig. 23, for vapor barrier *B*. As shown by the weight and in Fig. 22, when barrier *A* was used a slight amount of frost was collected on the aluminum panel, mostly at the bottom section. The inner surface of the sheathing did not show any signs of frost. When barrier *B* was used no frost was collected on either the aluminum panel or the inside surface of the sheathing. Thus this was an effective barrier to the passage of moisture.

The effect of adding a moisture barrier to the inside surface finish of a wall is shown in the results for walls Nos. 1-A, 3-A, 4-A, 7-A, 8-A and 24 of tests 3, second period. The amount of condensation ranged from 0.15 grams per 24 hours for one coat of asphalt paint to 0.33 for one coat size and two coats white flat paint. Two coats size, 2 coats aluminum paint and one coat of size with 2 coats of aluminum paint all showed 0.22 grams of condensate. It appears that a very effective barrier to the passage of vapor may be applied as an inside surface finish to the wall. However, more experimental work is needed to determine the best barriers to select.

The effect due to ventilating the sheathing to the outside has received only a small amount of attention thus far. In wall No. 5-A, test 3, two 1-in. vent holes were drilled through the sheathing and siding at the top and bottom of the center stud space. This resulted in zero condensation but it also lowered the surface temperature of the sheathing. In some of the walls on other tests ¼ in. holes were drilled and were found too small to be effective. It is evident that such ventilation may be established, but more investigation is required to find the practical limits.

RESULTS FOR CEILING CONSTRUCTION

The ceilings were of somewhat different construction than the side walls in that a 50 lb building paper was placed between the top of the joists and the outside sheathing. Three coats of white lead paint were applied directly to the sheathing with no other exterior finish. Some of the ceilings were constructed without insulation and others were constructed with different types of fill insulation. The construction of the ceilings is shown in Table 2; the comparative test results for those used in test 3 period 2 are shown in Table 7, and the conditions after certain tests are shown in the photographs, Figs 25 to 29, inclusive.

The building paper on the cold side of the insulation acted as a moisture barrier and collected much more frost than was collected on the sheathing of similarly constructed side walls. It appeared to be a much better barrier to the passage of moisture at low temperatures than it was at high temperatures. Thus in the uninsulated ceiling it was evident that some of the vapor which passed into the space between the joists was either escaping through the building paper which was at a fairly warm temperature or else it was

leaking through the joints around the edges of the ceiling. In wall No. 18, Fig. 19, in which a 50 lb building paper was used as a vapor barrier between the metal lath and studs, a considerable amount of vapor passed through the paper as shown by the weight of frost condensed on the inside of the sheathing. In ceiling C-2, Fig. 28, in which insulation was used, an aluminum sheet was placed between central joists with the building paper exposed in the two outer joist sections. After test 3, period 2, the building paper on the two outside joist sections was removed and weighed and then dried and re-weighed to determine the amount of frost collected. The amount was found to be exactly the same as that collected by the central aluminum panel, indicating that the building paper was a rather effective barrier at the low temperatures. This,

TABLE 7—RELATIVE CONDENSATION FOR DIFFERENT TYPES OF CEILING CONSTRUCTION UNDER SIMILAR TEST CONDITIONS

TYPE OF CEILING AND INSULATION	SPECIAL VAPOR BARRIER	CEILING NO.	PHOTO AFTER TEST FIG. NO.	TEST NO. AND PERIOD NO.	COLD ROOM F	SURFACE TEMP. OF SHEATHING	CONDENSATE GRAMS PER SQ FT PER 24 HOURS
Standard with 3⅝ in. rock wool	None	C-2	28	3–2	− 7.7	6.9	2.68
Standard with 3⅝ in. expanded aluminum magnesium silicate	None	C-1-A	..	3–2	− 7.7	9.2	2.45
Standard with 3⅝ in. redwood bark fiber	None	C-5	..	3–2	− 7.7	7.4	2.96
Standard with 3⅝ in. sawdust	None	C-6	..	3–2	− 7.7	9.6	2.45
Standard with 3⅝ in. rock wool	B[a] lapped joints	C-3	..	3–2	− 7.7	4.2	0.02
Standard with 3⅝ in. rock wool	B[a] sealed joints	C-4	29	3–2	− 7.7	6.8	0.15

[a] All vapor barriers placed between metal lath and studs.
B = Asphalt impregnated and surface coated sheathing paper, glossy surfaced and weighing 50 lb per roll of 500 sq ft.

however, cannot be taken as conclusive proof as some of the test data showed higher vapor travel through the outside stud section than through the center stud section, possibly due to lower temperatures in the outside exposed corner section of the wall.

Fig. 25 shows the typical condition of the paper on the under side of the sheathing of an uninsulated wall after any of the test periods. In the longer periods there was slightly more ice and frost formation around the edges than there was in the shorter test. The central sections were always clear, and the formation of ice and frost around the edges was apparently due to the cooling effect from the exposed corners. In all cases of either ceilings or side walls where frost was formed there were heavier formations along the surfaces next to the exposed corners, and in some cases it was heavier along the central section between the studs of the insulated walls with clear lines next to the studs. This formation was due to the fact that the heat transfer through

the studs was greater than through the insulated sections between the studs, therefore giving a slightly warmer section of the sheathing next to the studs with less chance for the formation of frost.

A photograph of ceiling No. C-2 shown in Fig. 26 after 460 hours test period is typical of the results obtained for all insulated ceilings and walls with no moisture barriers on the warm side. The heavier frost formation in comparison with the uninsulated walls is accounted for by the fact that the exterior surface is much colder for these walls than for the uninsulated walls, and therefore condenses more of the vapor out of the space between the studs, which in turn lowers the vapor pressure in this space and gives a greater vapor pressure drop through the inside surface, with a corresponding increase in the vapor travel through this section of the wall.

The values in Table 7 observed for test 3, period 2 show the relative amounts of vapor traveling through the plaster of ceilings with and without vapor seal construction for different types of fill insulation. In the first four ceilings four different types of insulation were used and no vapor seal applied. The amounts of condensation are substantially the same. In the last two, asphalt impregnated and surface coated sheathing paper with glossy surfaces was applied on the studs underneath the metal lath and for these ceilings the amount of condensate was practically negligible.

MOISTURE CHANGE IN INSULATION

In several of the tests the rock wool was weighed when installed and taken out and weighed immediately after the test to determine whether or not there was any absorption of moisture. The results for these weights are shown for several walls in Tables 3 and 4. While there might have been some small losses in the material between weighing periods, every precaution was taken to prevent this and in the majority of cases a decrease in weight is shown, and in no case is there an increase greater than what might be expected by experimental error. These tests show conclusively that the rock wool did not absorb moisture from the vapor in the air during the tests.

CONCLUSIONS

There is no doubt but that the condensation of moisture in walls is a troublesome problem with many types of construction under adverse conditions, and that in the past many difficult situations have arisen due to a lack of understanding of the true nature of the problem and to the fact that different building materials have been combined without regard to the effect of these combinations on the thermal and vapor properties of a wall. Since the temperatures of the various parts of a wall are controlled by the type and amount of insulation used and the vapor densities in the corresponding parts are controlled by the vapor barriers used, the transmission of heat and vapor through a wall must be considered together, and in most cases an intelligent combination of insulation and vapor barriers will eliminate any chance of trouble. A fundamental principle often overlooked is that a vapor barrier should be placed on the warm side and not on the cold side of a wall.

While there is still much research work to be done concerning practical vapor barriers and the limiting conditions under which they may be used, the

results obtained thus far prove the practicability of solving the problem by applying high vapor resistance materials to the inner section and low vapor resistance materials to the outer section of the wall. The problem is not to be feared after it is once understood. It is the intention to continue the investigation not only to study the individual vapor barriers and their limiting conditions but also to consider all of the problems involved in practical house construction by the use of large scale test houses within the cold room.

ACKNOWLEDGMENTS

This research program has been initiated by the *National Rock and Slag Wool Association* in cooperation with the University of Minnesota to make a thorough study of the elements found in the condensation problem and to determine the most practical solution from the builder's point of view.

The authors wish to acknowledge the cooperation and assistance of the Technical Committee of the *National Rock and Slag Wool Association* in the entire program. This Committee, consisting of Messrs. Jan S. Irvine, Chairman; W. W. Collins; E. W. McMullen; C. L. Neumeister; Earl Williams, and the late Dr. B. Townshend have given freely of their time and efforts to make this work possible.

The authors and the Committee acknowledge the valuable assistance and cooperation given by L. V. Teesdale, Senior Engineer, and his associates, of the Forest Products Laboratory, U. S. Department of Agriculture, Madison, Wisconsin. Mr. Teesdale has met several times with the Committee and has contributed liberally from the experience accumulated in an extended study of this problem at the Forest Products Laboratory.

DISCUSSION

L. A. HARDING (WRITTEN): The authors are to be congratulated for the clear exposition of the problems relating to condensation within walls. The paper serves the purpose of directing the attention of engineers to the various causes which are or may be contributary to this undesirable condition.

It will be apparent, after the reading of the paper, that the problem is not as simple as many have assumed, in the past. The pioneer tests described by the authors will doubtless be the forerunner of many similar tests by not only the authors but others as well.

E. C. LLOYD (WRITTEN): The authors have pointed out in the first paragraph of their paper that the problem of condensation may be found in cold storage buildings in any climate. This is indeed true as condensation may well be considered public enemy number one where low temperature insulation is concerned. Early cold storage buildings followed conventional methods of construction and employed building papers as sealing materials on the warm side of the structures. Also they used fill insulating materials such as mineral wool, wood shavings and sawdust between wood studs and joists. Condensation accumulated in such insulation spaces to the end that insulation value depreciated and in many cases mold growth attacked and destroyed the wood members.

The conditions prevailing in the wall of an air conditioned house in winter do not differ greatly from the conditions which prevail in the cold storage building in the summer season, in both cases high temperatures and high humidity on one side, low

temperature on the other. The cold storage building operator, after passing through a period of trying combinations of building papers and loose fills, found about 30 years ago that most satisfactory results in insulation were secured by the use of a homogeneous solid insulation moisture resistant in character, applied so as to provide continuous insulation which was sealed by application in hot asphalt against the penetration of moisture laden air. Now I point this out not so much as a suggestion as to a possible trend of insulation of air conditioned structures as to show that the idea of sealing on the warm side has been well and thoroughly considered for many years and perfection has not yet been reached.

The most serious condition to be met in the filled stud walls such as were tested in this paper is the accumulation of moisture in or on studs and sheathing rather than in the insulation itself. The results of moisture in a building structure where circulation is restricted are too well known to need amplification. Under the conditions of the test it is not to be expected that moisture would accumulate in the insulation material itself.

One question I should like to ask is regarding the pressure conditions which existed between the air in the test houses and the air in the room. As far as I can find there is no mention as to whether the pressure on the warm side was positive or negative. To me this seems highly important even though the pressure differences which prevailed may have been only a fractional inch of water.

Another question is regarding the condition of the mineral wool in the enclosing room. It would be most interesting to know whether any examination has been made of the insulating structure which encloses the outer room, as it would seem that an examination of this might reveal just as interesting data as an examination of the small inner rooms.

The authors are to be congratulated on the start that has been made on a real study of this condensation problem and it is to be hoped that the work will continue to develop more interesting data on the vapor-proofing of structures and the movement of moisture in building walls.

H. C. BATES (WRITTEN): Studying this excellent paper rather carefully, I note that stress was placed on the barrier method of preventing the accumulation of moisture or frost. This is a logical approach. I have made some conclusions from Table 6 which is reported as having results of tests which were made under the same conditions and are therefore directly comparable. I would like to have an opinion as to whether these conclusions are correct.

1. I note that there were four tests in which no condensation was found. These represent three cases sealed by special barriers on the warm side of the wall, and one case in which vents were provided on the cold side. Are those methods applicable to the walls of homes?

2. There are 11 tests showing condensation in the insulated panels less than that in the uninsulated; one, the same, and five greater. Do these 11 represent barriers and other conditions applicable to ordinary construction?

3. As mentioned in the text, there seems to have been a penetration of moisture through the sheathing on the uninsulated panel. A study of the vapor pressure and the amount of condensate indicates that about as much passed through the sheathing as was actually measured. It is to be expected that there will be penetration and escape through the sheathing and siding of an insulated panel when the inside sheathing temperature rises. In ordinary outdoor conditions, higher temperature will be reached during the day in most climates and, therefore, tend to reduce the amount of condensate even in walls with no moisture barriers. I have noted south side temperatures of 30 F during the day in

which the preceding night time temperature was −10 F and the succeeding night temperature +2 F, the north side temperature being about +21 F. Is it not reasonable to expect less condensation in houses than is shown under this test?

4. In order to show the magnitude of the condensation we have taken wall No. 18 which has building paper with lap joints between the metal lath and studs. Computing the excess moisture deposit for a house of average size, we find that there will be about 10 pints per month more than for an uninsulated house, if the relative humidity is maintained at 40 per cent and the outside temperature is −7.7 F. It is quite obvious that this moisture deposit is insignificant.

5. I understand that condensation is not troublesome in old houses which are insulated by blowing. Is this due to the porous structure which permits breathing to the outside or to the fact that high humidities are not maintained?

This work demonstrates that the condensation difficulty can be avoided so that those planning new houses may, without fear of trouble from condensation, take advantage of the comfort and saving which modern construction, including insulation, air conditioning and weatherstripping, has made available.

PROF. E. R. QUEER (WRITTEN): Mark Twain once commented in effect, that everyone talks about the weather but no one does anything about it. The problem of moisture and condensation in walls and insulating materials, likewise has been talked of a great deal in the past and no one did anything much about it. Fortunately for all of us concerned with insulation and building construction, Professor Rowley and his colleagues have made real progress in the solution of this annoying problem.

It is well to keep in mind that when adverse temperature and humidity conditions exist within walls, condensation will occur regardless of the type of insulation used. The problem is not new. The wood lath outlines on the finished plaster surface of a building that has been in service for a period of time illustrate the effects of condensation. The plaster key is believed to have a higher heat transfer value, consequently a lower temperature than the portion backed up by the wood lath and condensation occurs on the surface over the key causing a dark mark. The modern tendency to build insulated structures less impervious to air infiltration along with increased indoor humidities has served to amplify the general condensation difficulties. All these improvements are desirable and the responsibility for the difficulties should not be shifted to any particular one. The most logical solution seems to be to accent each factor as a necessity, make a wise choice of building materials, assemble them properly and place a good moisture barrier on or close to the inside surface of the structure.

As briefly outlined in the paper, it is proposed to investigate the extent of the moisture problem under normal and extreme conditions, measure the relative value of the various methods of correcting the difficulties and find the humidity conditions under which various types of building construction may be used without excessive condensation troubles and still retain the benefits of humidity and good construction. The apparatus described seems to be quite adequate to accomplish these investigations. However, it is hoped that the heat losses through the walls will be measured while moisture transmission progresses, and if possible, a correlation between these factors obtained. The writer appreciates the difficulties involved in making a correlation with transient conditions such as will be encountered but feels that the over-all heat losses from each box should be measured.

There are many phases of the problem which should be considered. A wall that may be satisfactory for the north-central section may not be suitable for the north-eastern section of the country. In the former case where hot dry conditions prevail for a time during the summer any accumulated moisture in the wall should be released. Whereas, in the latter section high summer humidities may prevent drying

out. It is true that a good wall should not accumulate any moisture but as soon as the outer wall protection is made to release the moisture, other problems arise. Care should be observed in applying low vapor resisting material to the outer section of the wall so that infiltration of moisture from hard driving rains such as are experienced along the Atlantic seaboard will be overcome. A heavy downpour lasting from 24 to 36 hours with a 45-mile gale blowing will saturate a wall unless the outer section is made very tight.

The results set forth in this preliminary report indicate interesting facts and show that a solution to the problem should be possible. The tests in a program of this nature should not be too highly accelerated. Time is an important factor and should be given due consideration. The results are expressed in moisture gain—grams per square foot per 24 hours for the test period under consideration. This gain is an average value and should not be interpreted to mean that it was constant from the beginning to the end of the particular test. The values should not be quoted to condemn any particular material as is so often done in tests of this nature. Later results may alter conclusions based on preliminary tests.

R. L. DAVISON: In hanging a picture, if you drove a nail into a wall that has this waterproof paper back of it and then took the nail out, what influence would that have on condensation in the wall? The reason that I ask this question is that some papers I have read by refrigerating engineers say that a small hole is just as bad as a large one when it comes to a question of moisture penetration into a wall.

PROF. F. B. ROWLEY: Mr. Lloyd refers to the similarity between the insulating problem in cold storage work and that in air conditioning buildings. It is true that many of the problems have been met in cold storage work, but it does not seem practical to use the same elaborate applications for air conditioned buildings as have worked out for cold storage plants.

Referring to Mr. Lloyd's question on pressures, the vapor pressures which seem to be the most important insofar as vapor transmission is concerned have been recorded. The static pressures due to the difference in air temperatures have not been recorded but might be calculated from the inside and outside air temperatures the same as for any building.

An examination was made to determine the condition of the insulating wool in the test room. The insulation was entirely dry excepting for a place where the ceiling joined the side walls in which there was not a break in the outside vapor barrier. The method of applying the vapor barrier on the warm side appears to be satisfactory providing there is an adequate continuous barrier.

The following comments apply to the five conclusions reached by Mr. Bates in analyzing test data:

1. It is practical to prevent condensation by placing a good vapor barrier under the metal lath on the warm side of the insulation. This method can only be applied to new construction. Venting a wall to the exterior will reduce the chances of condensation but may not in all cases be 100 per cent effective. In wall 5-A of Table 6 the vent holes at top and bottom passed through sheathing and siding. The insulation was not packed against the sheathing, thus there was reasonably good ventilation.

2. Those insulated walls which show condensation values to be lower than for the uninsulated wall are provided with vapor barriers on the warm side of the insulation. These are applicable to ordinary construction.

3. I agree with Mr. Bates that the test conditions are more severe than would be met in practice due to the warming up of the house during the middle of the day. As soon as the sheathing temperatures rise the rate of vapor transmission

from the wall to the exterior will be increased and reduce the rate of moisture of frost accumulation.

4. As stated by Mr. Bates the accumulation of frost under ordinary conditions is insignificant and it is usually only the extreme conditions which cause trouble.

5. There is some natural venting of walls through the exterior surface which varies greatly with construction. If condensation occurs it may be due either to poor ventilation or high relative humidities, or to a combination of both causes.

In reply to Mr. Davison's question I presume a nail hole would have some effect and no doubt if nail holes were placed at different elevations there would be a considerable amount of vapor circulation and the effectiveness of the barrier might be lost for extreme conditions.

MR. DAVISON: Have you made any tests of gypsum wall board in comparison to metal lath? Would the wall board act as a barrier?

PROFESSOR ROWLEY: No, we have not changed the interior construction. We probably will change the type of plaster base and go on and study those, but so far it has all been metal lath and plaster, no rock lath and no interior surface on the plaster base. I suppose that question might come up, too.

MR. DAVISON: Would you consider a gypsum wall board a barrier?

PROFESSOR ROWLEY: I am not prepared to pass on that. I think rock lath may have some benefit, but I am not sure if the paper does not act as just as much of a barrier. I do not want to pass on any material without testing it.

L. H. LAFFOLEY: I might give probably a slightly different angle to this question, and I am giving it as a suggestion for some of the future tests. If anything might be said of the present tests it is that they are probably too severe. In other words, if you live in a climate where you are going to get —17, the majority of houses are not built of wood. You get into much more solid types of construction, and I think you will find that the problem will be considerably varied from your wood constructed house. We do have wood houses in Canada undoubtedly, but where you get into the areas where it is colder and where you get longer periods of cold, you find your construction is a more solid and more permanent type. I think that angle should be considered if you are going to use as severe temperature as —17 as a basis for tests which you will formulate an opinion on.

L. K. HUGHES: Mr. Laffoley practically took up what I was going to say, except that I might mention that in the far northern mining district a tremendous amount of frame construction prevails. As the problem of condensation within walls seems to increase as outside temperatures decrease, this becomes increasingly important from the Canadian standpoint especially in these areas, where temperatures of —45 and —50 deg may prevail for many consecutive days, and where there is bright sunshine for several hours per day on the same walls. What effect might such conditions have on this problem?

PROFESSOR ROWLEY: Well, of course, the longer the prolonged period of cold the more condensation, but the point you bring up of sunshine on a wall I think relieves this problem. I do not doubt but what 20 below zero for 20 days is a pretty severe condition even in Minnesota, and even during that period you have a lot of sunshine. I believe that sunshine on a wall would reduce the severity of the condition and probably take away a lot of that moisture that has accumulated. That will be a part of our program from now on after we set up the maximum conditions.

B. F. MCLOUTH: I would like to ask Professor Rowley what happens to the frost as the temperature outside rises. Does the frost condense into liquid and run down

the surface of the sheathing, or does it evaporate directly back into a vapor and pass harmlessly outside of the partition?

PROFESSOR ROWLEY: I think it does both. It will depend on how fast it evaporates. If you had no sudden evaporation I think it would all evaporate and pass out and you would not know it was there, but if you had a sudden warming up so fast that it would thaw, I think it would run down the wall. I think you can get either condition depending upon the heat that is applied.

J. U. MALEY:[2] I would like to ask Professor Rowley if he has given any consideration to steel studs and the effect of condensation on the inside surfaces of the walls with those?

PROFESSOR ROWLEY: We have not made any tests of steel studs, but naturally a steel stud in a wall gives you a good path for heat transfer, the same as any other high conductivity part of the wall, and the general principle is to avoid any high conductivity paths from surface to surface of a wall if you want to keep the temperature of the inside surface warm and in good condition. We have not actually done any work in this study on steel studs.

H. L. STEVENS: A question was brought up before in regard to condensation. If that wall was rock wool, would it not have a tendency to hold that condensation in the wall without drying out?

PROFESSOR ROWLEY: Rock wool is not hygroscopic, that is it does not absorb moisture directly from the air to any great extent. If it should become soaked with water as a sponge I see no reason why it would not dry out the same as any other material under similar conditions.

MR. STEVENS: I am presuming your rock wool is damp or wet. How does it get wet without leakage into the wall?

PROFESSOR ROWLEY: The insulation may become wet by the condensation of vapor which penetrates to the inner surface of the wall and comes in contact with materials which are below its dew-point temperatures. This may happen to any material through any part of a wall under certain conditions of vapor pressures and temperatures.

MR. STEVENS: The point I am bringing up mostly is they say rock wool will not dry out again. Will it dry out again?

PROFESSOR ROWLEY: There is no reason why rock wool will not dry out the same as any other material.

J. H. BRACKEN: I have just one question. Professor Rowley has mentioned the interior heat stop and also venting on the external wall. You regard those as alternates, do you not? You do not advocate the use of both?

PROFESSOR ROWLEY: Well, I don't know that they would be alternates. You might use one, a vapor stop, on the inside in one case and a vent on the other side. It is relative, you see. You are preventing it from passing in.

MR. BRACKEN: You do not recommend both in the same wall?

PROFESSOR ROWLEY: There would be no objection as I see it if you had a condition in which you wanted to use both, if you could get your conditions best by using both vapor stop and a vent, but practically every material does pass some vapor, so the question of an absolute stop or an absolute conductor of vapor depends on the relative values. You might have a material, and you probably do in all cases, on the outside which will pass some vapor, but it is a question of the relative amount that

[2] Housing Division, Truscan Steel Co., Youngstown, Ohio.

passes in and passes out of the wall. I do not see any objection to using both a stop on the inside and a vapor release valve on the outside.

MR. BRACKEN: But if one is efficient, you do not see why we should use both?

PROFESSOR ROWLEY: No, if you stop the vapor on the inside I do not think you would need venting on the outside.

No. 1078

STUDIES ON BACTERIAL CONTROL IN AIR CONDITIONING

By T. S. CARSWELL,* H. K. NASON,** AND J. D. FLEMING *** (*NON-MEMBERS*)
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THE efficiency of the air-washer in removing micro-organisms from circulating air has long been of interest to engineers and sanitarians, but the literature reporting the results of workers in this field is scanty, confused and lacking in conclusive evidence.

The research reported in this paper constitutes an extension of previous studies. In all this work, the aim has been to consider an air-washing unit as a *machine for removal of micro-organisms.* No attempt is made to touch upon other phases of the problem, such as the eventual effect upon the bacterial population of the air conditioned space, or upon the possible effect upon human health. It is believed that the studies reported in this paper clear up most of the confusion regarding washing of micro-organisms from air, and present a rational picture of the mechanism involved in bacterial removal by air-washing.

In a previous paper [1] Carswell, Doubly and Nason have reviewed the literature relating to bacterial control in air conditioning. This review need not be repeated here except to point out a few particular facts to furnish the necessary background. While it is frequently assumed that water scrubbing of circulating air, as practiced in the average air-washer or humidifier, will eliminate bacteria and molds as well as dusts and pollens, there is actually much evidence to the contrary.

The Whipples [2] are apparently the only other workers on record who reported a reduction in the bacterial content of air after water washing, and their results varied widely from 0 to 100 per cent. Larsen,[3] on the other hand, in investigating the efficiency of an air-washer in a school building, found that "when using recirculated water the washer supplied bacteria to the air instead

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[1] Bacterial Control in Air Conditioning, by T. S. Carswell, J. A. Doubly and H. K. Nason (*Industrial and Engineering Chemistry, 29,* 85, 1937).

[2] Some Preliminary Studies in Air Washing and Its Results, by G. C. Whipple and M. C. Whipple (*American Journal of Public Health, 3,* 1138, 1913).

[3] Tests on the Recirculation of Washed Air, by G. L. Larson (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916, p. 213).

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1938, by H. K. Nason.

of removing them, and even when using new water continuously, it did not show any marked efficiency as a bacteria remover." Clark and Gage [4] showed that the use of air-washer water polluted with *B. coli* resulted in an increase in the number of these organisms in the humidified air of textile mills. Wells [5] showed that in air-washing systems using polluted water, the bacteria of the water were thrown into the air and could be recovered from distant parts of the building. In some cases, most of the bacteria present in humidified air were introduced from the humidifying water.

In 1936, Carswell, Doubly and Nason found that washing effected some reduction in the bacterial content of circulating air and that the addition of a germicide, benzylphenol, to the water gave a much better reduction. It was also found that when sufficient germicide was used in the washing water, the latter was maintained in a practically sterile condition, whereas it rapidly became contaminated when no germicide was added; most, if not all, of the organisms coming from the air being washed.

The investigation herewith reported was undertaken in an attempt to:

1. Obtain fundamental data on the efficiency of the air-washer as an apparatus for bacterial purification of air;
2. Investigate more completely the effect of adding germicide to the washing water; and
3. Formulate a clearer concept of the mechanism of air purification by water washing.

It should be emphasized that this work was directed solely toward obtaining data on the bacterial effectiveness of air-washing equipment and that no conclusions of medical or hygienic significance can or should be inferred.

APPARATUS AND METHODS

Three separate air conditioning systems were studied which were similar in design. Each system was arranged with an air-washer of the *high efficiency* type in which humidity and temperature control was obtained by washing the air with sprays of chilled water. The systems were installed in a pharmaceutical manufacturing plant, a theater and a general office building, all of which were located in the city of St. Louis.

The study covered summer operation only, and while local atmospheric conditions varied somewhat, cooling and dehumidification were generally being effected. The bacterial and fungal contents of both air and water were studied, and a complete record was kept of all pertinent physical and mechanical data.

The micro-organism content of the air was determined with the aid of an air centrifuge [6] which has been found to give the most accurate results of any method yet devised. Samples of air were drawn through sterile tubes from the air ducts and passed through the instrument. The micro-organisms present were deposited by centrifugal force on a layer of culture medium which

[4] Annual Report, by Clark and Gage (Massachusetts Department of Health, 1912).
[5] Effect of Polluted Water for Purposes of Humidification, by W. F. Wells and E. C. Riley (Massachusetts Department of Health, Annual Report, 1934, p. 166).
[6] Apparatus for Study of the Bacterial Behavior of Air, by W. F. Wells (*American Journal of Public Health, 23*, 58, 1933).
IV Report of Sub-Committee on Bacterial Procedures in Air Analysis: A Bacteriological Method of Sanitary Air Analysis, by W. F. Wells, E. B. Phelps and C.-E. A. Winslow (*American Public Health Association* Year Book, 1936-1937, p. 97).

was supported on the walls of a revolving culture tube. Upon suitable incubation, a visible colony developed on the medium at each point where a living micro-organism had been deposited. The air velocity through the sampling tube was measured continuously by an orifice tube and differential manometer. By sampling for definite times, the sample volumes could be calculated accurately. Corrections for temperature and pressure were made.

Air samples were taken from the main air ducts just prior to entering the washing chamber and several feet beyond the circulating blower on the effluent side of the washer. By comparing the micro-organism content of these samples, the effect of washing could be readily determined. Samples of air-washing water were collected from the circulating pump and micro-organism contents determined by standard methods.[7]

By using various culture media and incubation temperatures, the air-borne micro-organisms were roughly separated into the following groups:

1. Bacteria growing at body temperature (possibly parasitic or pathogenic);
2. Bacteria growing below body temperature (probably not parasitic nor pathogenic but possibly of industrial importance); and
3. Fungi.

Bacteria were determined by incubation on nutrient agar at 37 C for 48 hours or at 26 C for 72 hours. Fungi were determined by incubation on Czepek's agar at 26 C for 72 hours. Some fungi also appeared on the nutrient agar cultures at both temperatures and were counted separately. The general procedure adopted was to run complete tests for a period of one week under normal operating conditions with no germicide in the wash water. The same tests were then run the following week under conditions as nearly identical as possible except that germicide was maintained in the washing water.

Any germicide used in air conditioning should be highly potent against the micro-organisms involved, non-corrosive to metal used in the equipment, odorless when used in effective concentration, non-volatile, non-toxic to man and to higher animals, economical and safe to use, stable even on prolonged aeration and easily dissolved in water. A commercial mixture of the ortho- and para-benzylphenols meets these stringent requirements and was used throughout this work. The benzylphenol was dissolved by adding the theoretical equivalent of sodium hydroxide; this was necessary because the free benzylphenol is sparingly soluble in water while the corresponding sodium salt (phenate) is quite soluble.

A benzylphenol concentration of 500 parts per million has been found to be the optimum for insuring sterilization of the water and this concentration was maintained throughout the test period by addition as needed of a stock solution of the chemical. Additional tests have shown that the use of higher concentrations, up to 1000 ppm, gave no further reduction in bacterial content of either air or water. The amount necessary was determined periodically by chemical analysis, using a rapid volumetric method, and additions regulated accordingly. By this method, the desired concentration could be closely controlled. Sodium hydroxide solution was also added as needed, to maintain a phenolphthalein alkalinity of about 20 ppm (expressed as $CaCO_3$), thus insuring against precipitation of the germicide by CO_2 from the air. The alkali also

[7] Standard Methods of Water Analysis (*American Public Health Association*, 8th Edition, 1936).

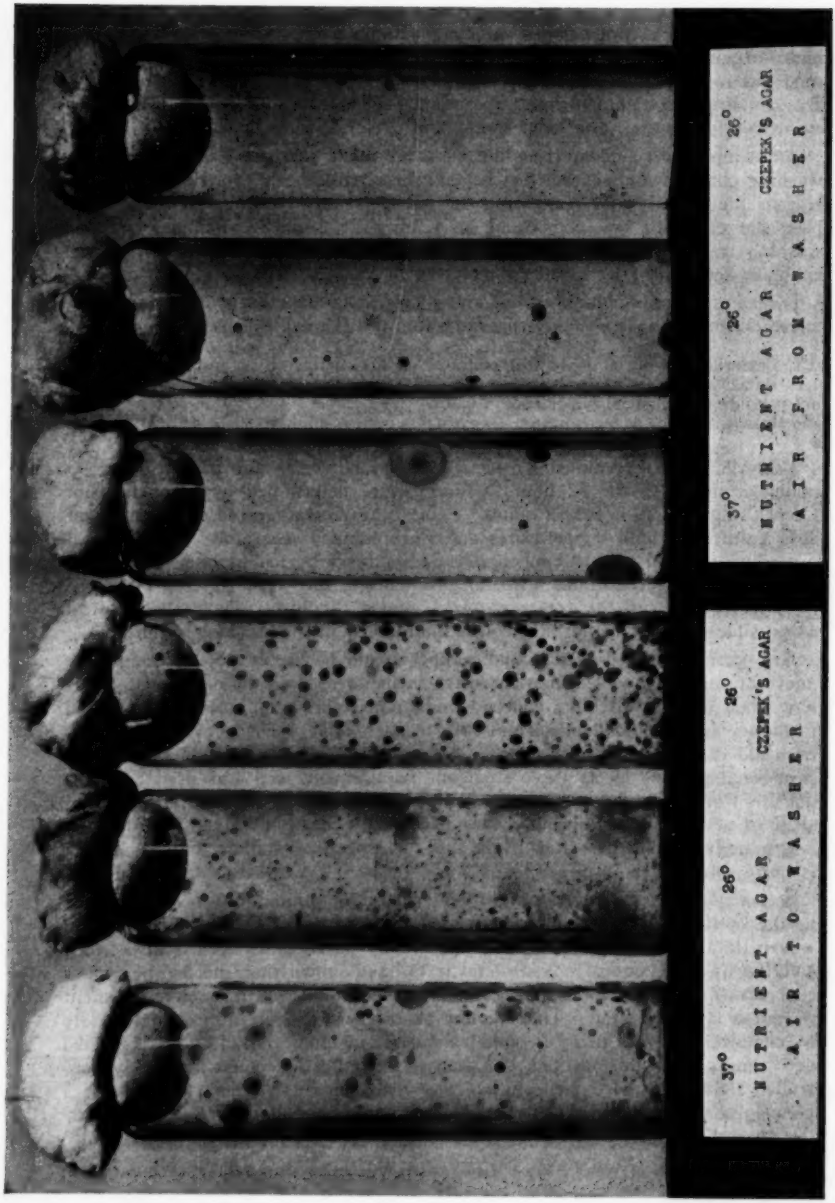

FIG. 1. BACTERIA AND FUNGI PRESENT IN RECIRCULATED AIR, 500 PPM BENZYLPHENOL IN WASHING WATER

guarded against corrosion by regulation of *pH* (hydrogen-ion concentration) and hence was of additional value.

The water was changed and the washer cleaned thoroughly before each series of tests. Complete records of temperatures, air velocities and other physical factors were kept for each set of tests.

RESULTS OBTAINED

In all, 526 separate determinations of the micro-organism content of air and 84 determinations of water pollution were made. The data obtained are so extensive that they can only be summarized in this paper.

TABLE 1—AVERAGE 37 C BACTERIAL AND FUNGAL RELATIONSHIPS (NUTRIENT AGAR)

MICRO-BIOLOGICAL DATA		INSTALLATION		
		Office Building	Theater	Pharmaceutical Plant
Bacteria per cc water, no germicide		100	1203	5400
Bacteria per cc water, 500 ppm benzylphenol		3.2	1.2	12.7
No Germicide	Organisms per 100 cu ft air to washer	269	475	359
	Organisms per 100 cu ft air from washer	54	125	277
	Reduction, organisms per 100 cu ft washed air	215	350	82
	Per cent reduction	80.3	73.7	22.8
500 ppm Benzylphenol	Organisms per 100 cu ft air to washer	388	585	511
	Organisms per 100 cu ft air from washer	76	81	103
	Reduction organisms per 100 cu ft washed air	312	504	408
	Per cent reduction	80.6	84.9	78.1
Improvement in per cent reduction due to use of germicide		0.3	11.2	55.3

The results obtained when air was washed with water containing germicide are given in Fig. 1. This photograph shows centrifuge-culture tubes for one set of tests upon the air to and from the washer, which in this case was that installed in the theater. The first tube from the left should be compared with the fourth, the second with the fifth, the third with the sixth, to show the relative microbial content of the air samples before and after washing. The efficient removal of organisms by the air-washer is apparent.

Tables 1, 2 and 3 summarize the micro-biological data developed in these tests. The specific determinations for each test period have been averaged to simplify comparison.

These tables represent data which have been highly condensed from a large number of observations. Table 1 averages both bacterial and fungal relationships as determined by incubation at 37 C on nutrient agar. Table 2 averages fungal relationships only as determined by incubation at 26 C on Czepek's agar. Any bacterial growth shown by the latter plates was dis-

TABLE 2—AVERAGE 26 C FUNGAL RELATIONSHIPS (CZEPEK'S AGAR)

	MICRO-BIOLOGICAL DATA	INSTALLATION		
		Office Building	Theater	Pharmaceutical Plant
No Germicide	Fungi per 100 cu ft air to washer	883	1771	142
	Fungi per 100 cu ft air from washer	83	146	64
	Reduction, organisms per 100 cu ft washed air	800	1625	78
	Per cent reduction	90.8	87.0	57.3
500 ppm Benzylphenol	Fungi per 100 cu ft air to washer	495	2079	303
	Fungi per 100 cu ft air from washer	49	101	51
	Reduction, organisms per 100 cu ft washed air	446	1978	252
	Per cent reduction	90.7	95.0	83.9
Improvement in per cent reduction due to use of germicide		− 0.1	8.0	26.6

regarded; in fact, only a few bacterial colonies were present, as most bacteria cannot utilize this medium. Table 3 averages both bacterial and fungal colonies incubated at 26 C on nutrient agar. An analysis of fungal and bacterial counts in the case of both Tables 1 and 3 showed that the per cent reduction was approximately the same for either type of growth, and closely equal to the average figures presented in the table for the combined organisms.

A more detailed study of the individual data disclosed the fact that the bacterial efficiency of a washer was nearly constant during the test periods when germicide was used. In the periods when no germicide was used, the

TABLE 3—AVERAGE 26 C BACTERIAL AND FUNGAL RELATIONSHIPS (NUTRIENT AGAR)

	MICRO-BIOLOGICAL DATA	INSTALLATION		
		Office Building	Theater	Pharmaceutical Plant
No Germicide	Organisms per 100 cu ft air to washer	1295	2783	936
	Organisms per 100 cu ft air from washer	176	561	342
	Reduction, organisms per 100 cu ft washed air	1119	2222	594
	Per cent reduction	85.7	80.0	65.6
500 ppm Benzylphenol	Organisms per 100 cu ft air to washer	876	3323	712
	Organisms per 100 cu ft air from washer	159	330	167
	Reduction, organisms per 100 cu ft washed air	717	2996	545
	Per cent reduction	80.8	90.0	74.4
Improvement in per cent reduction due to use of germicide		−4.9	10.0	8.8

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efficiency was initially high and was nearly that maintained when germicide was used, but declined steadily and rapidly as the test progressed. The decrease in efficiency was much more marked in those cases where the bacterial pollution of the wash water increased markedly. In one case, when the wash water never became highly polluted, no marked decrease in bacterial efficiency of the washer could be noticed. Figs. 2 and 3 show this effect clearly.

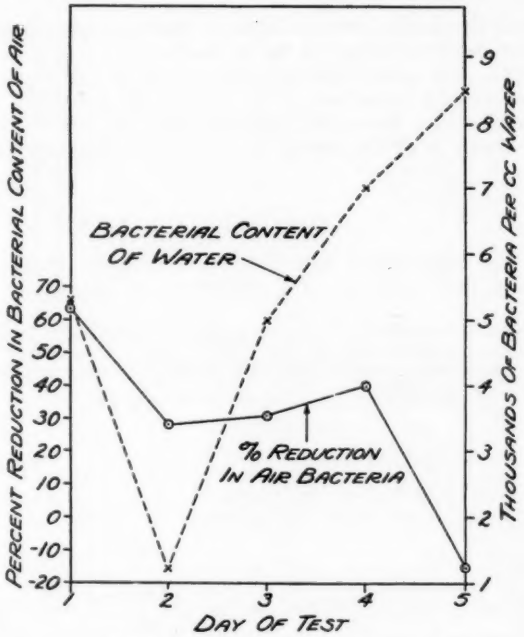

FIG. 2. PHARMACEUTICAL PLANT. RELATION BETWEEN BACTERIAL POLLUTION OF WASHING WATER AND BACTERIAL EFFICIENCY OF AIR WASHER (48 HOURS INCUBATION AT 37 C)

The effect of germicide in the washing water is to render the water sterile and when germicide is used, the bacterial efficiency of the air-washer tends to remain nearly constant. In general, the use of germicide, while maintaining base efficiency does not increase the efficiency beyond that obtained with naturally sterile water. In other words, the germicide acts on the microorganisms only in the water phase. This is logical when it is considered that air particles are in contact with the germicidal sprays for a fraction of a second only,—too short a time for the germicide to kill. When the organisms are taken into the water phase, however, they may remain in contact with the germicide for several minutes and thus be readily destroyed by the chemical.

The efficiency of the washer will thus be limited only by failure of the

organisms to be mechanically taken into the water phase, either because the sprays do not completely cover the cross-sectional area of the washing chamber, or because part of the air may pass through the sprays without being wetted.

The available evidence also confirms the conclusions of other workers that when highly polluted water is used bacteria may be furnished to the air rather than removed from it. This is shown in Fig. 2 by the negative value for per cent reduction in bacterial content.

That a direct relationship between degree of micro-biological pollution of the washing water and the efficiency of the air-washer with respect to removal of micro-organisms does exist is clearly demonstrated by Fig. 4, in which data from all three systems studied are plotted. It will be noted that the increase in bacterial efficiency of the washer when the water was sterilized with germicide varies directly with the degree of contamination of the water before sterilization.

CONCLUSIONS

Based on the experimental data obtained, a rational picture of the air washing process may be drawn. The conclusions are as follows:

1. When operated on sterile water, an air-washer possesses a certain base efficiency in removing micro-organisms from the air. This efficiency is apparently constant for the particular system (under uniform operating conditions) and is probably a function of the design of the equipment.

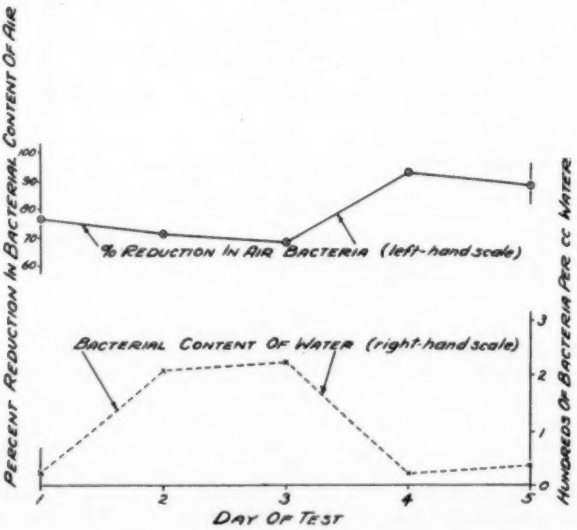

FIG. 3. OFFICE BUILDING. RELATION BETWEEN BACTERIAL POLLUTION OF WASHING WATER AND BACTERIAL EFFICIENCY OF AIR WASHER (48 HOURS INCUBATION AT 37 C)

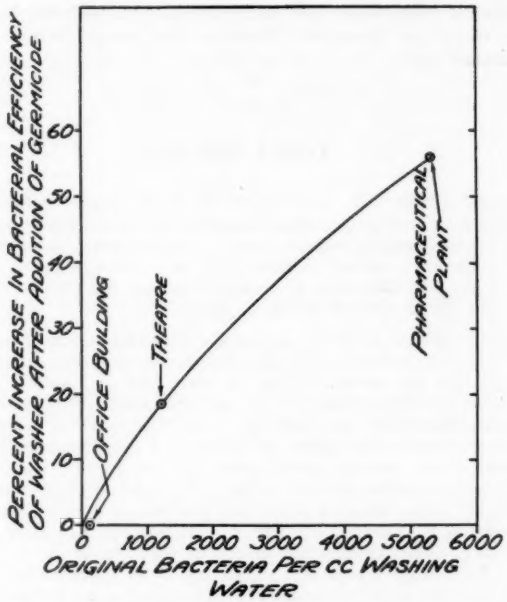

FIG. 4. RELATION BETWEEN BACTERIAL POLLUTION OF AIR-WASHING WATER AND IMPROVEMENT IN BACTERIAL EFFICIENCY OF AIR WASHER OBTAINED BY STERILIZATION OF WATER WITH GERMICIDE (48 HOURS INCUBATION AT 37 C)

2. There is a direct relationship between the degree of microbial contamination of the air-washing water and the efficiency of the air-washer as an apparatus for removing micro-organisms from the air.

3. As the degree of pollution of the washing water increases, the microbial efficiency of the air-washer decreases, and if the pollution becomes sufficiently heavy, the washer efficiency becomes negative. That is, it actually furnishes organisms to the air rather than removing them from it.

4. The addition of sufficient germicide to the washing water to render it sterile insures continuous operation at base efficiency, but does not seem to increase the efficiency beyond this range. The use of germicide concentrations higher than those required to maintain sterility in the washer water does not increase the efficiency of the air-washer with respect to microbial reduction.

5. Bacteria and fungi are affected similarly by the washing process.

6. An air-washer operated on sterile water is an efficient apparatus for bacterial control of air, and the use of a suitable germicide is a convenient way to insure sterility of the washing water.

ACKNOWLEDGMENTS

The authors are indebted to L. L. Lewis and J. H. Bailey of the Carrier Corp. for their aid in securing permission from the owners of the equipment to make these tests. Staffs of the several organizations operating these units

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rendered valuable assistance and their splendid cooperation is appreciated. R. S. Shumard of the Monsanto Chemical laboratory assisted with some of the bacteriological tests.

DISCUSSION

V. T. KARTORIE (WRITTEN): Aside from the thermodynamic advantages of spray type dehumidifiers over surface cooling equipment, this paper establishes the efficiency of an air washer in removing bacteria from the air stream. The conclusions of the authors are admittedly a rational picture of the air washing process, nevertheless on the strength of the data presented, the results bear out the fundamental concepts of the reduction of bacteria with sterile water sprays.

It is somewhat difficult to have a complete and thorough understanding of the performance of a dehumidifier with and without the use of a germicide, when a great many test runs are summarized as has been done in Tables 1, 2, and 3. For example, in all three of these tables the per cent reduction of bacteria in the air phase for the pharmaceutical plant was markedly less either with or without the use of these germicides. In the water phase too, Table 1, it would appear at a glance that the dehumidifier in the pharmaceutical plant was the worst. But this is not the case. If we were to compare the bacteria per cc of water in the washer pan, with the germicide with the count when no germicide was used, it is found that the ratio for the office building is the lowest $3.2 \div 100$ or approximately 1 to 31.3; in the pharmaceutical plant $12.7 \div 5400$ or approximately 1 to 427; and in the theater $1.2 \div 1203$ or 1 to 1000. This simple comparison shows that if the dehumidifier in the pharmaceutical plant was the most inefficient as would appear from the per cent reduction in the air phase, the same is not necessarily true in the washer pan.

As brought out by the authors, the degree of the contamination of the washer water before sterilization affects the efficiency in the air phase.

Also it is evident that in order to obtain effective bacterial control in the air phase, not only the washer water has to be treated, but a dehumidifier of high humidifying efficiency must be used. It would seem logical to expect that when the air velocity is relatively low, and a large quantity of water is sprayed per square foot of cross-sectional area, a high reduction in bacterial count in the air leaving the dehumidifier would result.

According to the authors the optimum concentration of the germicide has been found to be 500 ppm, and the tests with 1000 ppm resulted in no further reduction. Then it is logical to conclude that the concentration of 500 ppm gives the highest efficiency, and that lower concentrations would probably result in lower efficiencies. Just how low the concentration may be dropped is very important from the application standpoint. It seems that this study is the logical problem for further research.

After the washer studies have been completed it would be desirable to determine the eventual effect on the conditioned spaces. It is acknowledged that the authors did not attempt to consider this; however it must be remembered that the bacterial reduction in the conditioned spaces is of primary importance, because the customer is more interested in what conditions will result in the spaces than how the dehumidifier produces them.

Figs. 2 and 3 are well done, and are very conclusive as they are presented. Perhaps it would be more desirable to plot per cent reduction and bacteria count in relation to hours of operation rather than in relation to day of test. When calculating the cost of bacterial treatment it would be necessary to know the operating

time in hours during which the germicide is effective, and when the washer water would have to be changed.

The authors are to be complimented on the excellent manner of presenting their paper. They have shown that bacterial control is possible in air conditioning systems employing spray type dehumidifiers. Further study and development of this phase of air conditioning may alter the designs of systems for hospitals, hotels, and public buildings where bacterial contamination is frequently a problem.

H. C. MURPHY (WRITTEN): These studies on bacterial control in air conditioning are a definite contribution to progress in this field.

The fact is frequently overlooked, however, that most of our present day air conditioning equipment eliminates a large percentage of bacteria and similar air-borne organisms.

In a recent Air Pollution Survey by Dr. Leon Buchbinder of the Columbia Medical Center which is reported in the April 1937 issue of *The American Journal of Public Health*, it was found that in air conditioning systems the air filters, installed to remove dust, were also removing a large percentage of bacteria.

Their surveys extended over a period of 18 months and covered a number of schools, theaters, subways, parks, etc.; they found spaces supplied with filtered air from air conditioning units to have definitely lower bacteria counts than untreated areas.

A theater, for instance, in the Rockefeller Center district, equipped with ordinary commercial air filters had a bacteria count 83.4 per cent lower than an unventilated theater. The bacteria count in the air conditioned theater was in fact slightly less than in the park.

Dr. C. B. Coulter and Dr. F. M. Stone of the College of Physicians and Surgeons, Columbia University, similarly reported in the October 1937 issue of *The American Journal of Public Health* that they found greatly reduced bacterial counts in telephone booths supplied with filtered air over non-ventilated booths.

For some years it has been possible to remove 95 to 99 per cent of air-borne bacteria by the use of suitable air filters.

Since 1922 much of the better grades of gelatine has been produced under filtered air and records show an average removal of in excess of 98 per cent of air-borne organisms.

A midwestern manufacturer of surgical sutures has since 1928 maintained practically sterile air conditions in his manufacturing processes by bacteria air filters.

A manufacturer of nut butter reduced the mold spore count in his factory 99.6 per cent by suitable air filtration.

A number of the larger pharmaceutical manufacturers conduct many of their manufacturing processes under filtered air and report highly effective removal of air-borne organisms.

Practically all modern breweries utilize air filters to eliminate bacteria and certain mold spores which cause difficulties in the brewing processes.

A more recent but very promising method of bacteria control in air conditioning is that suggested by Prof. W. F. Wells in the use of the ultra-violet ray. There is need for much additional research in this entire field.

H. L. BAKER (WRITTEN): While it is essential to condense materially the substance of observed data when preparing a paper of this nature, it ocurred to the writer that incorporation of additional information such as water temperature curve on Figs.

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2 and 3 would have been very interesting. Since these were systems under automatic control and used refrigerated water, it is not improbable that water temperatures may have varied as much as 20 to 25 deg during the test period. The effect of this variation on the mortality and multiplication rate of bacteria probably accounts for the decrease in water pollution during a test when one would normally expect a continued increase. In spite of the fact that Fig. 3 establishes the conclusion that there is a direct relationship between the microbial contamination of the air washer water and efficiency of the air washer for removing these micro-organisms, it appears that some reason should have been advanced for the inconsistency of Fig. 2. It is apparent that in this test some condition, untabulated in this paper, is exerting a more powerful influence on the microbial efficiency of the washer than is the bacterial content of the washing water. While this paper did not attempt to extend beyond the apparatus operation, it is evident that it has opened up additional fields for investigation; *first,* the possible use of germicides other than benzylphenol which may be utilized; *second,* the automatic control of germicide concentrations in washer water; *third,* the utilization of this principle of washer operation to the advantage of both industrial and comfort air conditioning. As an illustration of this last point, it certainly does not take much imagination to consider a fourth item on the program of the Technical Advisory Committee on Hospital Air Conditioning, such as the utilization of recirculated air following sterilization during washing operation.

A. A. ADLER: I have no comments to make on the testing, because apparently the authors have done a very fine job; nevertheless I am reminded of the history of antiseptics. As a boy, drug stores used to sell a lot of it. Barber, butcher and other shops used the word antiseptic in their appeal to customers.

The question arises as to why all germs found in water and air should be killed. Do we know which are the harmful types of germs and in what numbers they ought to be present? Unless we know what we are after, I do not see that we have a definite outline of the problem. The mere fact that we can kill germs in large numbers is interesting to know, but wholesale slaughter has been a part of the history of civilization. Everything we misunderstand we wish to destroy.

A good example of a fine research work that is going on is in the electron field. Electrons, as you know, have no personalities; they are all alike. We study one particular element, find its reaction with other elements in the problem, and build synthetically to the conclusion.

The problem in air conditioning is not as simple as that of electronics. Many obscure reactions are taking place. The objective is as yet indefinite.

Physicians are not sufficiently able as physicists to understand what engineers are trying to do in air conditioning. Engineers are not sufficiently good physicians to know what they ought to be doing under the circumstances. However, we have equipment and talent at our command.

Our first step should be to formulate the problem. With a correct specification before us the solution is a matter of logic. We have the talent available in our and other societies. The commercial application will then pass through the usual evolutionary stages.

F. R. BICHOWSKY: I merely want to ask these two questions. Were any measurements made on the two engineering efficiencies of the washer? I mean the efficiency in respect to water removal and the efficiency in respect to the removal of sensible heat. It would be very interesting indeed to correlate the efficiency of bacterial removal with the efficiency of the removal of the two engineering factors in the design of washers. It may be that the problem of the control of bacteria is, as was suggested, a purely design problem, a problem of the design of the efficiency of

the washer, but if so we should know that because that would make another very good reason for the design of still more efficient washers.

The second point I wish to bring out is the need of including a third factor in statistical studies of this type. The removal of molds and of various types of bacteria is, of course, important, but to the practical engineer it is also important to have data on the efficiency of the washer to remove the spores, plant spores, because of the hay-fever bugaboo.

H. K. NASON: With regard to Mr. Kartorie's comments: his calculation of ratio of bacterial content before and after washing is very interesting and possibly of some significance. He suggests that further work be done to determine the minimum concentration of germicide applicable. Such a study is being made. In preliminary tests we found that when the concentration is reduced below 400 ppm, bacterial pollution in the water begins to increase. Hence, we have reason to believe that 400 ppm can be considered the minimum effective concentration. Believing it desirable to include a small factor of safety, we have used and are recommending a concentration of 500 ppm for tests with benzylphenol. Further experiments are being made to determine more precisely the minimum effective concentration, but we consider 500 ppm to be a safe, desirable and easily maintained concentration, and one which gives practical sterility.

Mr. Murphy has given a very good discussion on other methods of bacterial removal which is, of course, essential to any consideration of this problem. However, we would like to cite certain advantages which the use of a germicide in the air-washing water possesses over other methods.

In the first place, the use of germicide in the washing water achieves control of micro-biological slimes which are very prone to form in air-washing equipment and especially in inaccessible parts of the equipment where mechanical cleaning does not reach.

The second advantage, which is closely related to the first one, is the control of odors produced by micro-biological slimes.

It should be desirable to use germicide in addition to air filters, to provide a second line of defense against bacteria, since there is a possibility that air filters may lose their efficiency on prolonged use.

The ultra-violet method of sterilization is of extreme interest, particularly to the medical field, and it has been used with considerable success in operating rooms. One may, however, question the practicability of this method of sterilization for use if it is installed in the ducts of an air conditioning system where the humidity is generally at or near saturation, since Professor Wells' results indicate that ultra-violet light may be less efficient under high humidity conditions. All of these methods of control, namely air filters, ultra-violet light and the use of germicide in the wash water, are important and interrelated and all of them should be considered as useful and valuable methods where bacterial or fungal control is desired.

Mr. Baker has raised the question of the effect of water temperature on bacterial content of the air-washing water. This point was settled to our own satisfaction by taking a sample of air-washer water under sterile conditions, sealing it into a sterile glass bottle, and suspending this in the actual air-washing pan for a period of one week. From a theoretical standpoint it would be expected that no multiplication of bacteria would occur at the low temperature usually present in air-washing water, and this test offered confirmation of this theory because no multiplication did occur. The bacterial content of the sealed sample remained practically constant while the bacterial content of the water circulating in the system increased considerably.

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The criticism of Fig. 2 as being inconsistent with Fig. 3 is well taken. If in Fig. 2 we consider the trend of the two curves the inconsistency is not as striking as it will appear to be if we consider only actual points.

Mr. Adler has raised the question of significance, which we have been very careful to avoid. We do not feel that we are in a position to discuss hygienic significance, and I believe that it is no affront to members of the engineering profession to say that they are not in a position to discuss it. This question has been considered thoroughly by the workers at Harvard who are properly qualified. Professor Wells believes, according to his published statement, that while we cannot prove that it is desirable from a medical standpoint to remove bacteria from the air, it is certainly a good idea to do so. The condition is much the same as in the purification of water, and the discussion that has arisen is much the same. It may do no harm to use polluted drinking water, but since it is so easy and safe to keep it from being polluted, why not do it?

Entirely aside from the medical significance of micro-biological purification of air, the industrial importance is worth consideration. There are many industries which need or require germ-free air, and any method, whether chemical or physical, which will tend to provide more desirable micro-biological conditions for these industries is worth consideration.

Mr. Bichowsky asks if we have made measurements on the efficiency of the washer with respect to water removal or removal of sensible heat. These data have not been tabulated, but this can be done, as the necessary operating information is available. We do have actual water removal measurements and theoretical water removal measurements (made by tracing the loss of salt from these systems), and we find that those two measurements check one another very closely.

The question of removal of plant spores is also mentioned. While no direct measurements were made on these, I think it is safe to say from a theoretical standpoint that they will be removed by any means that will remove bacteria, since they are so much larger and so much more easily removed than bacteria. We do know that they are removed by air filters which do not remove bacteria very efficiently. This conclusion is found in the published works of Professor Drinker and Professor Wells at Harvard.

No. 1079

PHYSIOLOGIC RESPONSE OF MAN TO ENVIRONMENTAL TEMPERATURE

By F. K. Hick, M.D., Ph.D.,* R. W. Keeton, M.D.** and Nathaniel Glickman, M.S.*** (*NON-MEMBERS*), Chicago, Ill.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of Illinois, College of Medicine

THE response of the body to atmospheric environment has long been a subject of interest, which has filled literature with important observations. In the earlier studies investigators have taken advantage of fortuitous circumstances, which constituted conditions for a controlled experiment. Thus Hunt,[1] from studies in hot dry climates, made observations on the quantity of water which must be evaporated to render the body comfortable. Haldane,[2] working under conditions of high humidity, fixed the temperature (88 F expressed as a wet-bulb), at which the dissipation of heat was inadequate and the body developed a fever. Here also should be noted the numerous observations upon heat stroke.[3, 4] It is obvious that the outlook for progress from studies of this type was limited, since chance only rarely sets up conditions for a controlled experiment.

Another approach to this subject (Petersen,[5] Mills[6 a,b,c,d]) is found in the study of the effect of climate and weather on the individual. Since it is often difficult to measure quantitatively the adjustments of normal individuals to these changes, patients with disease were selected. The course of a given disease, with its periods of easily discernible invasion and remission, has been charted.

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*** A.S.H.V.E. Research Fellow.

[1] The Regulation of Body Temperature in Extremes of Dry Heat, by E. H. Hunt (*Journal of Hygiene*, 12:479, 1912).
[2] The Influence of High Air Temperatures, by J. S. Haldane (*Journal of Hygiene*, 5:494, 1905).
[3] Physiological Aspect of Heat Stroke, by L. Hill (*Lancet, London*, 1:661, 1920).
[4] Thermal Adjustment of Man and Animals to External Conditions, by Sir C. J. Martin (*Lancet*, London, 2:673, 1930).
[5] The Patient and the Weather, by W. F. Petersen (Edwards Bros., Ann Arbor, Mich., 1934-37).
[6a] Climatic Factors in Acute Nephritis, by C. A. Mills (*Journal of Hygiene*, 16:871, 1932).
[6b] Climate as a Potential Factor in the Etiology of Exophthalmic Goitre, by C. A. Mills (*Endocrinology*, 16:52, 1932).
[6c] Acute Appendicitis and Weather, by C. A. Mills (*Journal of Medicine*, 15:30, 1934).
[6d] Climatic Basis for Susceptibility to Heat Stroke or Exhaustion, by C. A. Mills and Cordelia Ogle (*American Journal Hygiene*, 17:686, 1933).

Presented at the 44th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1938, by Dr. F. K. Hick.

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The weather records covering the same period have been studied and the variable factors traced. Where a given weather variable has shown a close correlation with the swings of disease, it has been considered significant in modifying the body's response. This important type of observation has already served to focus attention on factors which must be subjected to experimental study.

For an experimental approach to the problem, the capacity of the environment to transmit and dissipate heat should be known or measurable. Hill [7] introduced the Kata thermometer for this purpose, and it was pointed out by Winslow [8] that changes in this instrument are an approximate index of bodily comfort. Somewhat later other studies (Houghten [9]) led to the development of the *effective temperature* scale, which is now in general use. This scale integrates the effects of temperature, humidity and air velocity. Winslow and associates [10] have now arrived at an expression for the radiation properties of the environment. It would seem therefore that in the near future some agreement will be reached as to the definition and analysis of the heat dissipating capacity of the environment. Until such is obtained, workers will continue to speak different languages and interpret their results diversely.

When one turns to the behavior of the human body, factors are found which render the problem more complex. The oxidative processes develop heat, and the body dissipates it, thus modifying the environmental heat capacity. This constitutes an effect of the body on its environment. The body, in turn, must lose a given amount of heat, or the heat will accumulate and elevate the body temperature, thus producing a fever. The machinery used for this purpose is complex. The methods employed vary widely, but they are conditioned by the capacity of the atmosphere for heat dissipation. Under one set of conditions the body's heat may be lost entirely by evaporation and under other conditions chiefly by radiation. In these adjustments an effect of the environment on the body is produced.

The problem of an individual's adjustment to environment is essentially one in physiology, and must be approached from this experimental viewpoint. DuBois [11] and Winslow et al.[12 a,b] have recently analyzed quantitatively the forms of heat loss of the body for a rather wide range of temperatures and humidity. The large role played by circulation in making it possible for the body to lose heat under a variety of circumstances has become apparent to all workers. Under one set of conditions the skin and periphery is warm and moist, and under another dry and cool. At times the pulse rate is speeded up, and the individual complains of overaction of the heart. On a hot day people are driven to drink increased quantities of water by thirst, a physiologic device

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[7] Physiology of Open Air Treatment, by L. Hill (*Lancet*, 184:1285, 1913).

[8] The Kata Thermometer as a Measure of the Effect of Atmospheric Conditions Upon Bodily Comfort, by C.-E. A. Winslow (*Science*, 43:716, 1936).

[9] Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yagloglou (TRANSACTIONS A.S.H.V.E., 29:163, 1923).

[10] Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal Physiology*, 120:1, 1937).

[11] The Mechanism of Heat Loss in Health and Disease, by E. F. DuBois (*Transactions Association of Physicians*, 51:252, 1936).

[12a] The Relative Influence of Radiation and Convection Upon Vasomotor Temperature Regulation, by L. P. Herrington, C.-E. A. Winslow and A. P. Gagge (*American Journal Physiology*, 120:133, 1937).

[12b] Physiological Reactions of the Human Body to Various Atmospheric Humidities, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal Physiology*, 120:288, 1937).

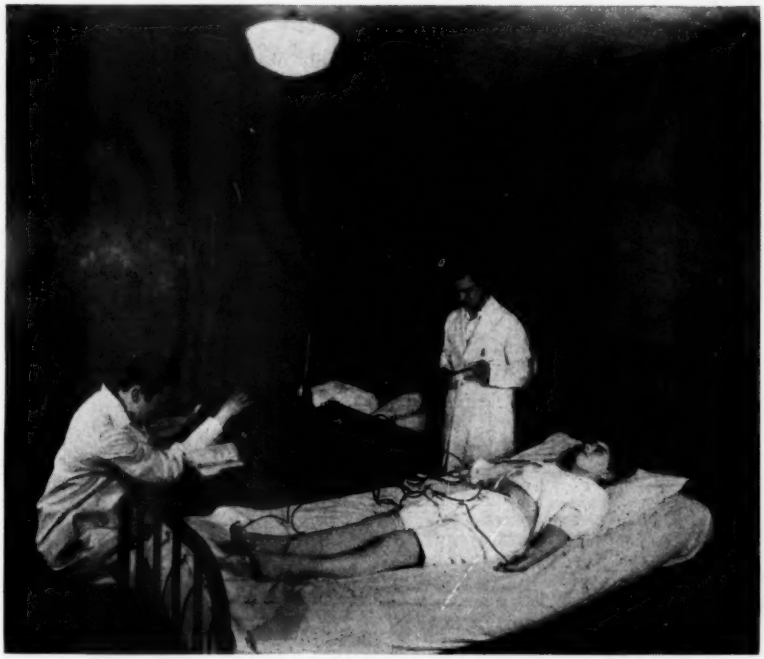

FIG. 1. SKIN TEMPERATURE MEASUREMENTS ON SUBJECT IN AIR CONDITIONED ROOM

designed to replenish blood volume. These and similar observations are familiar to all. An analysis of such adjustments must be made. This includes changes in heart action, blood vessel behavior, blood volume, and blood distribution. In the present study are reported certain objective measurements, which are to be regarded as preliminary to a major objective, the adjustment of the circulation to the environment.

EXPERIMENTAL DETAILS

Subjects. This study was performed with the assistance of the house officer staff at the Illinois Research and Educational Hospital located in Chicago. Since normal subjects are scarcely available for hospitalization for weeks at a time, patients were used having certain complaints which required hospitalization and yet had no abnormality in heat regulation which could be detected. It is felt that the responses of these subjects are essentially those of normal subjects. The subject I.W. was a healthy male of 24, a medical student. The other subjects had been hospitalized for some time, were ambulatory, and were psychologically suited to the restriction of activity required of them. They were sufficiently trained in the procedures used to make their responses

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reliable. The subjects are designated by initial, sex mark, age and diagnosis on the illustrations.

Observations. The variables on which daily observations were made were: basal metabolism or heat production, vital capacity, skin temperature and rectal temperature, and the presence of sweating or shivering.

Procedure. The air conditioned room is a small chamber accommodating two beds on the medicine floor of the Illinois Research and Educational Hospital. The conditions within the room were checked daily at several places and different times by a sling psychrometer. Air velocity was measured by a vane type anemometer recording an air movement of about 25 fpm. The fans were operated at a constant speed.

Two subjects dressed in standard cotton pajamas entered the room in the evening and made themselves comfortable under suitable bed clothing. About

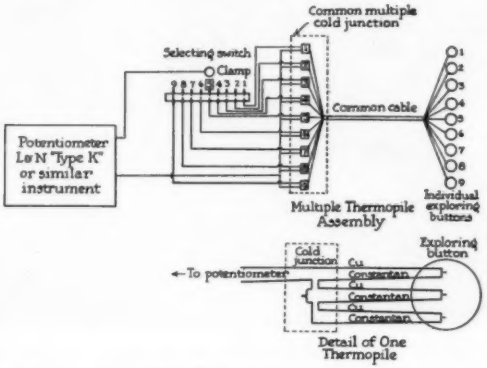

FIG. 2. DIAGRAM OF CIRCUIT OF ELECTROMOTIVE THERMOMETER

6 a.m. they were awakened, the covers thrown back and they lay quietly for 1 to 2 hours before measurements for that day were begun (Fig. 1). In order to standardize costume the subjects rolled up the sleeves and legs of the pajamas as shown and had the trunk largely exposed.

First, two measures were made of the heat produced per unit time, basal metabolism. Then skin temperature measurement was carried out, and the rectal temperature taken. Finally the vital capacity was measured.

Skin temperature measurements were made by means of a thermopile arrangement having several exploring or unknown temperature junctions. The electrical circuit is diagrammed in Fig. 2. The cold junctions, each consisting of 3 copper constantan junctions, were kept beside a mercury thermometer. Copper wire was wrapped about them all, then they were introduced into a test tube which was kept in a thermos jar filled with ice and water. The exploring button was patterned after the apparatus of Sheard.[13] The circuit differed from Sheard's in having a different switch and in the use of a potentiometer as a

[13] The Electromotive Thermometer: An Instrument and a Method for Measuring Intravenous, Superficial and Cavity Temperatures, by C. Sheard (*American Journal Clinical Pathology*, 1:209, 1931).

null point instrument rather than using a deflection galvanometer. The exploring buttons were held on the skin by narrow strips of adhesive. This

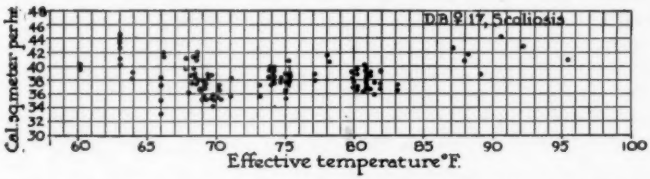

FIG. 3a. CORRELATION OF CALORIES PER SQUARE METER PER HOUR AGAINST EFFECTIVE TEMPERATURE

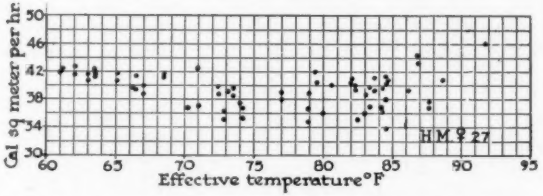

FIG. 3b. CORRELATION OF CALORIES PER SQUARE METER PER HOUR AGAINST EFFECTIVE TEMPERATURE

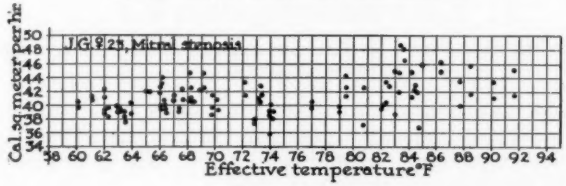

FIG. 3c. CORRELATION OF CALORIES PER SQUARE METER PER HOUR AGAINST EFFECTIVE TEMPERATURE

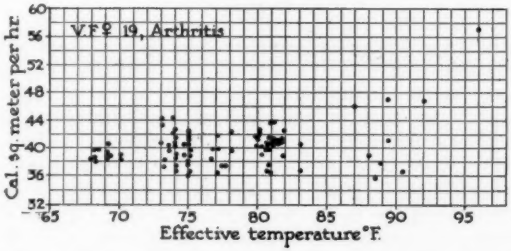

FIG. 3d. CORRELATION OF CALORIES PER SQUARE METER PER HOUR AGAINST EFFECTIVE TEMPERATURE

apparatus carries the usual probable errors of similar devices. It depends on contact with the skin, yet may by contact insulate the skin and so alter the temperature and/or alter the capillary circulation in the skin under the button.

Daily readings of the skin temperature were made with the buttons on the same points; forehead, sternum, balls of both middle fingers, the right forearm, balls of both great toes, and a point above the right patella.

Rectal temperature was taken by an ordinary clinical thermometer.

Basal metabolism, the heat produced per unit time expressed as calories, per square meter per hour, requires that the subject be rested by a night's sleep, have had no food for over 12 hours and be relaxed. The Tissot tank method was used. In this procedure the expired air is collected, its volume measured, and its content of CO_2 and O_2 determined by analysis. This analysis permits the calculation of the respiratory quotient (R.Q.). The R.Q. is $\frac{CO_2 \text{ expired}}{O_2 \text{ inspired}}$. If the R.Q. is at wide variance from the normal, the situation is not *basal* and the results meaningless. A mouth piece and nose clip were regularly used.

The vital capacity is defined as the maximum volume of air a subject can exhale after the deepest breath he can take. A simple spirometer measures

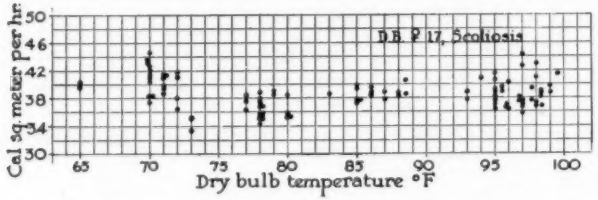

FIG. 4. CORRELATION OF CALORIES PER SQUARE METER PER HOUR AGAINST DRY-BULB

it. Although very different in the different subjects, it is relatively constant for each individual. Although a few other factors control vital capacity, it largely mirrors the amount of blood in the lungs. The greater the volume of blood in the lungs, the less air they can hold.

Sweating and shivering were sought for by inspection and recorded daily.

RESULTS

Heat Production. Basal metabolism is recorded on the charts in Fig. 3, where the effective temperature is the abscissa and the heat production expressed in calories per square meter per hour is the ordinate. (A calorie per square meter per hour is equivalent to 0.369 Btu per square foot per hour.) In such distribution charts there is a scatter of points on either side of a trend line. If the heat production were a function of the effective temperature, some correlation should be apparent at a glance. If perfect correlation were obtained, all the points would lie on a single line. The points are least widespread when effective temperature is the abscissa. When the dry-bulb (Fig. 4) is used on the abscissa, the scatter is a bit wider, though not greatly different. Similarly, the basal metabolism is not a function of the wet-bulb.

Fig. 3a shows an apparent slight rise in the heat production at the cool end of the curve. In this subject shivering was noted when effective temperatures below 65 deg were obtained. However, the rise in the curve does not prove

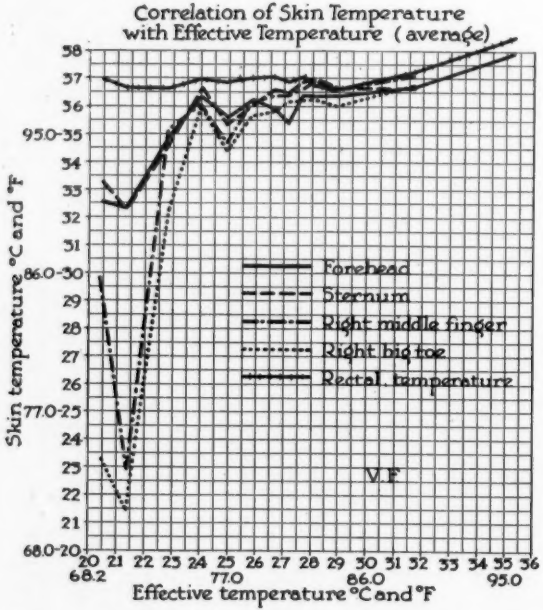

FIG. 5a. CORRELATION OF SKIN TEMPERATURE AGAINST EFFECTIVE TEMPERATURE

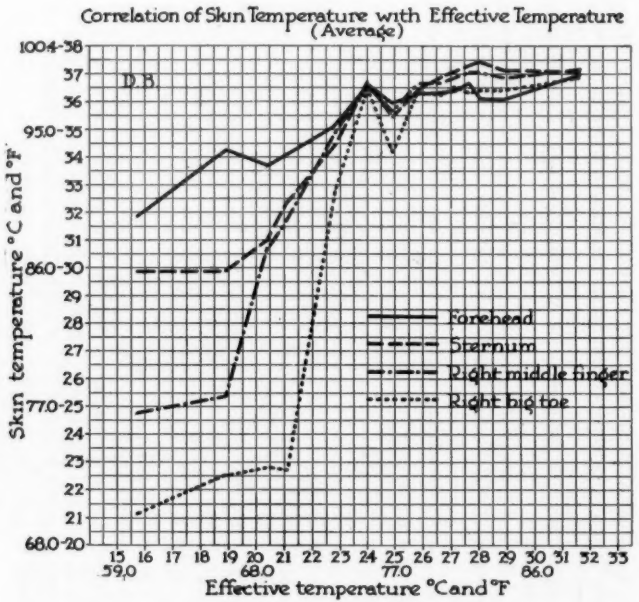

FIG. 5b. CORRELATION OF SKIN TEMPERATURE AGAINST EFFECTIVE TEMPERATURE

significant on statistical analysis of this (and all the other) curves. The variations observed in the basals were too great to permit this slight rise to bear significance.

Fig. 3b illustrates also the slight rise in heat production with shivering in a cold atmosphere.

Fig. 3c, J.G., a woman of 23 who had survived severe heart failure, did not show any rise in metabolism in the cooler ranges. She did not shiver. It

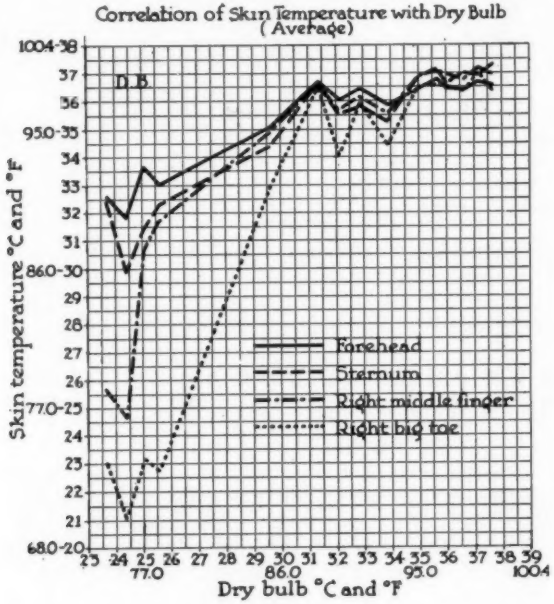

FIG. 6. CORRELATION OF SKIN TEMPERATURE AGAINST DRY-BULB

seems that her basal metabolism and body temperature began to rise at environmental temperatures lower than that of normal subjects.

Fig. 3d does not extend low enough to involve shivering.

All of these measurements were determined with the subjects at *basal* conditions. Under these conditions the heat produced becomes minimal for that individual.

Rubner [14] measured with a calorimeter the heat production of a dog kept at different dry-bulb temperatures. In the cold zone, below 16 C the animal's heat production rose to values greatly in excess of those observed at temperatures from 20 to 30 C. Fever was observed with temperature above 35 C in the environment. The *chemical regulation* of body temperature was distinguished from the physical regulation.

[14] Die Beziehungen der atmospharischen Feuchtigkeit zur Wasserdampfabgabe, by M. Rubner (*Arch. f. Hyg.*, 11:208, 1890).

There has been some dispute as to the existence of a chemical regulation of heat in man. Loewy,[15] among others (Johansson[16]), concluded that shivering and other movements might lead to increased heat production. In the absence of shivering, the heat production was unchanged. Hill[17 a,b] came to the opposite conclusion.

McConnell and Yaglou[18] did basal metabolic rate determinations on healthy men, and a curve was developed plotting basal metabolism against effective temperature as the abscissa. This curve rises at either end, and the rise diagrammed in the hot range is similarly suggested in the data reported in this paper. Winslow et al.[19] comment that the metabolism remains constant throughout the range of experimental temperatures observed. The subjects had food two hours before the determination was started. Digestion is associated with a variable rise in heat production of over 10 per cent.

The rise in either end of the temperature ranges studied by McConnell and Yaglou may partly result from the inability of subjects to remain quiet and relaxed when distressingly chilled, or acquiring a fever. In the one instance shivering is involuntary; in the other the subjects are increasingly restless. Experience acquired in this study supports this opinion. It is established that when fever appears, heat production rises in accord with van't Hoff's law.

The literature does not mention any man capable of lowering his heat production as the environment becomes too warm under acute conditions.

Data accumulated by animal experiments cannot be transferred to man when they deal with heat regulation. Mammals as a rule are insulated with hair, cannot sweat, or do so in small degree. They cannot readily vary their heat loss by radiation as man does. Therefore, mammals require some chemical regulation of heat production which man has replaced by better means of heat loss (sweating and variable radiation).

Data obtained in these observations support the contention that chemical regulation of body temperature in man plays an insignificant role within the ranges shown.

RESULTS OF SKIN TEMPERATURE MEASUREMENT

Skin temperature results are illustrated by the graphs in Fig. 5, where the abscissa is effective temperature. It is apparent that in the cooler zone the extremities, particularly the hands and feet, cool down, even being observed lower than room temperature. The temperature of the hand rises when the environment is somewhat warmer. When still warmer environment is present, the feet rise in temperature, reaching a peak about blood temperature, with an effective temperature of 75 deg. The temperatures of the sternum and forehead, upper arms and thighs diminish as the temperature of the environment falls. They do not show variations comparable in extent with the hands and

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[15] Ueber den Einfluss der Abkuhlung auf den Gaswechsel des Menschen, by A. Loewy (*Pfluger's Arch.*, 46:189, 1890).

[16] Ueber die Tagesschwankungen des Sloffwechsels und der Korpertemperatur in nüchtenem Zustande und vollständiger Muskelruhe, by J. E. Johansson (*Skandin. Arch. für Physiol.*, 7:123, 1897).

[17a] Report on Ventilation and the Effect of Open Air on the Respiratory Metabolism, by L. Hill (Local Gov. Bd. Report, New Ser., No. 100, 1914).

[17b] Observations on the Resting Metabolism of Children and Adults in Switzerland, by L. Hill and J. A. Campbell (*British Medical Journal*, 1:385, 1922).

[18] Basal Metabolism as Affected by Atmospheric Conditions, by W. J. McConnell and C. P. Yagloglou (*Archives of Internal Medicine*, 36:382, 1925).

[19] Loc. Cit. See Note 10.

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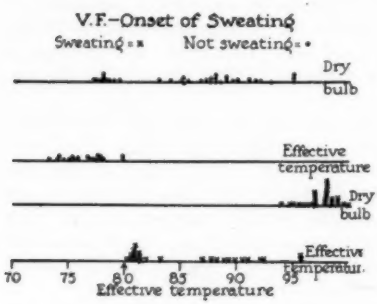

FIG. 7a. CORRELATION OF SWEATING AGAINST EFFECTIVE TEMPERATURE

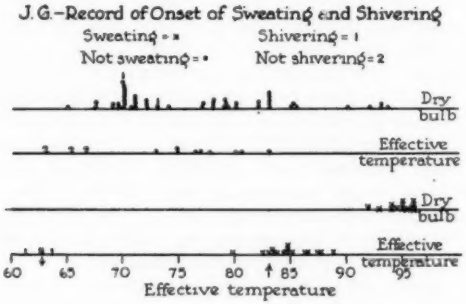

FIG. 7b. CORRELATION OF SWEATING AGAINST EFFECTIVE TEMPERATURE

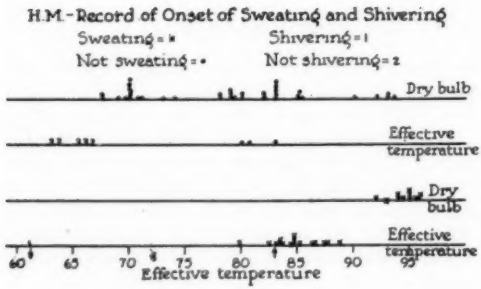

FIG. 7c. CORRELATION OF SWEATING AGAINST EFFECTIVE TEMPERATURE

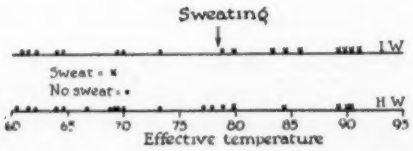

FIG. 7d. CORRELATION OF SWEATING AGAINST EFFECTIVE TEMPERATURE

feet. Above 82 deg ET the extremities continue to rise slowly, even exceeding finally the sternum and forehead.

At 77 deg ET there is observed a transient fall in temperature of the extremities, especially the hands and feet. Since this is slightly below the point at which sweat appears grossly on the skin, it is considered to be a change in the radiation of heat, while imperceptible perspiration comes out on the skin and assists with body cooling.

The work of Maddock and Coller [20a,b] and Sheard [21] established the variable radiation capacities of the extremities. Barbour [22] demonstrated the reflex regulation of skin temperature. Hardy [23] has measured the great sensitivity of the skin to application of radiant heat.

It is clear that a mechanism exists which regulates skin temperature so as to control heat loss by radiation. In the zone 66.0 to 72.5 deg ET this control rests largely in the hands. In the higher range, 70.0 to 75.0 deg ET, it is in the feet.

That these data can be best correlated with the effective temperature is not surprising. Fig. 6 indicates the less regular distribution of points when skin temperature is plotted against dry-bulb. It is well known that the amount of heat lost from the body by radiation depends on the temperature and character of the walls; that the heat lost by evaporation depends on the wet-bulb, the dry-bulb, air velocity and type of air movement; and the heat lost by conduction and convection on dry-bulb and air motion. It was anticipated that the effective temperature in combining three of these variables would come nearer correlation with physiologic observations.

SWEATING

The onset of sweating in these subjects (Fig. 7) was about 81 deg ET. This was rather constant where no disease process was significant in control of heat loss. The fact that these subjects were completely at rest and fasting permitted making this observation of the critical temperature for appearance of sweating. If the body's heat production should increase in response to exercise or the specific dynamic action of food, the onset of sweating would be at lower temperatures.

SHIVERING

The onset of shivering was not indicated at any critical temperature. In general it was observed below 65 deg ET.

VITAL CAPACITY

The results of vital capacity measurements are shown with scatter charts using effective temperature as the abscissa (Fig. 8). There is a consistent and

[20a] Role of the Extremities in the Dissipation of Heat, by W. G. Maddock and F. A. Coller (*American Journal Physiology*, 106:589, 1933).

[20b] The Function of Peripheral Vasoconstriction, by F. A. Coller and W. G. Maddock (*Annals of Surgery*, 100:983, 1934).

[21] Effect of Changes in Environmental Conditions on Skin Temperatures and the Dissipation of Heat from the Body, by C. Sheard, M. D. Williams and B. T. Horton (*Proceedings American Physiological Society*, p. 147, 1937).

[22] The Heat-Regulating Mechanism of the Body, by H. G. Barbour (*Physiological Review*, 1:295, 1921).

[23] Studies in Temperature Sensation III. The Sensitivity of the Body to Heat and the Spatial Summation of End Organ Responses, by J. D. Hardy and T. W. Oppel (*Journal of Clinical Investigation*, 16:533, 1937).

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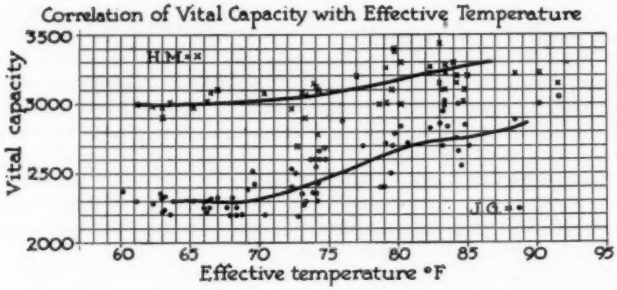

FIG. 8a. CORRELATION OF VITAL CAPACITY AGAINST EFFECTIVE TEMPERATURE

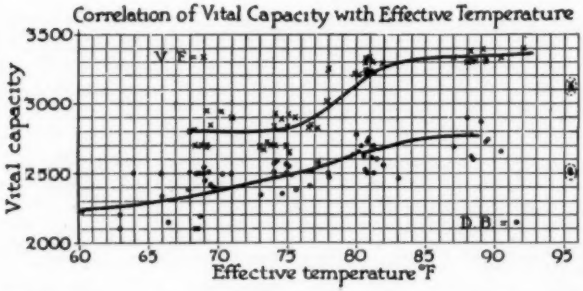

FIG. 8b. CORRELATION OF VITAL CAPACITY AGAINST EFFECTIVE TEMPERATURE

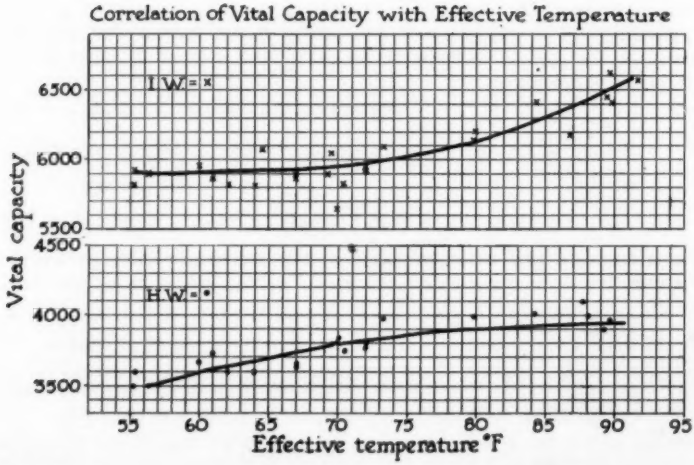

FIG. 8c. CORRELATION OF VITAL CAPACITY AGAINST EFFECTIVE TEMPERATURE

significant rise in vital capacity as the temperature rises. The rise exceeded 1000 cc in I. W. a healthy medical student and 800 cc for J. G. a girl with organic heart disease. The subjects were not aware of any change in sensations as their vital capacity rose. On the days when conditions were between 70 and 80 deg ET they expressed the opinion that their vital capacity would be higher. This prediction was regularly in error.

It is the authors' opinion that this rise in vital capacity reflects a shift in the distribution of blood within the body. An increased volume of blood in the skin would lead to a reduced quantity somewhere else. Blood reservoirs in the lungs [24 a,b,c,] and abdomen could supply this blood. The work of Moore and Kinsman [25] included measurement of the volume of blood within the chest and observed results which are supportive of the idea that the lungs normally act as a reservoir for storage of blood.

HYPERVENTILATION

Hyperventilation was noted as a response of some subjects to hot environments. Trained subjects, exposed to very high ET, breathe out per unit time a great deal more air than at lower ranges. This response is often given by nervous, and untrained individuals at lower temperatures. It is a common source of error in observations on untrained subjects against which suitable precautions must be taken. It was observed in these tests when the body temperature was rising.

When hyperventilation lasts over 2 min there is enough CO_2 washed out of the body to give rise to symptoms of headache, nausea, weakness, giddiness and prostration. In fact hyperventilation tetany is described with numbness and tingling in the extremities followed by characteristic muscle spasm.[26 a,b] This may be one of the mechanisms of producing discomfort in high environmental temperatures.

CONCLUSIONS

1. The *chemical* regulation of body temperature is negligible in man under basal conditions and within the ranges investigated.

2. Radiation of heat is a principal means of heat loss in a range below 80 deg ET. It is controlled by a sensitive regulation of skin temperature in the extremities.

3. The point at which sweating appears for a subject under basal conditions is about 81 deg ET. A drop in skin temperature of the extremities occurring

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[24a] The Effect of Pulmonary Congestion on the Ventilation of the Lungs, by C. K. Drinker, F. W. Peabody and H. L. Blumgart (*Journal Experimental Medicine*, 35:77, 1922).

[24b] Failure of the Circulation, by T. R. Harrison (Williams and Wilkins, Baltimore, p. 131, 1935).

[24c] Architectonics of Vessels of Lungs with Special Consideration of Blood Storing Function of Lungs, by R. A. Pfeifer (*Ztsch. f. Kreislaufforsch*, 26:906, 1934).

[25] Studies on the Circulation: Further Analysis of Injection Method and of Changes in Hemodynamics under Physiological and Pathological Conditions, by W. F. Hamilton, J. W. Moore, M. M. Kinsman and R. G. Spurling (*American Journal Physiology*, 99:534, 1932).

[26a] A Study of Forced Respiration. Experimental Production of Tetany, by S. B. Grant and A. Goldman (*American Journal Physiology*, 52:209, 1920).

[26b] The Effect of Prolonged Hyperpnea on the CO_2 Combining Power of the Plasma, the CO_2 Content of Alveolar Air and the Excretion of Acid and Basic Phosphate and Ammonia by the Kidney, by J. B. Collip and P. L. Bachus (*American Journal Physiology*, 51:658, 1920).

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a few degrees below that suggests that evaporation from the body surface has increased at 77 deg ET.

4. Under basal conditions rectal temperature does not rise until the effective temperature reaches 91 deg.

5. Evidence is presented of a redistribution of blood as the skin temperature rises, leading to a diminution of blood content of the lungs.

6. Hyperventilation is a transient response to intolerable heat.

7. The effective temperature scale comes nearest to correlation with the observations reported in these investigations. If to this could be added the factor of radiation capacity of the environment, better correlation could be expected.

DISCUSSION

J. J. AEBERLY (WRITTEN): The authors have clearly indicated in this report that their investigation should be considered in the light of pointing to their major problems rather than as a basic understanding underlying the phenomena of the transfer of heat from man to his environment. In this sense, I accept this report and will not make a critical examination of the data submitted. The authors have presented several interesting aspects of the problem and have pointed out the direction in which they believe studies should be made.

Of the several interesting questions indicated in connection with their findings, especially intriguing to me, is that a thought may arise in our minds concerning the data submitted in connection with vital capacities and presented in graphical form on Figs. 8a, b and c.

You will note a consistent rise in vital capacities due to rises in effective temperature. In the case of I. W. the increase has been approximately 12 per cent and in D. B.'s case the increase is 27 per cent, without any appreciable increases in the metabolic rates.

Physiologists tell us that the depth of respiration is regulated by the CO_2 content of the blood and that this control is extremely sensitive to changes of carbon dioxide concentration in the blood.

I wonder if the authors have given thought to this phase of the problem and have an explanation for it. Does this change of the CO_2 content in the alveoli, due to dilution and no increase in the metabolic rate, affect the CO_2 content of the blood which in turn controls the depth of respiration? If it does, why is this control in the reverse direction from expectation? It would seem that, if the vital capacity remained constant for any length of time at this greater volume, the partial pressure of CO_2 in the blood would be materially reduced because of the reduced partial pressure in the alveoli and ultimately must reflect itself in a lower CO_2 concentration reaching the medulla. This line of reasoning may answer at least a part of the question uppermost in the minds of many of our air conditioning engineers, namely, "What is the complete physical picture of the physiological response of man to the conditions of his air envelope?"

E. J. RODEE (WRITTEN): The results reported in this paper emphasize the necessity of finding all the physiological reactions of the human body to its environment before complete comfort standards can be established. While these results cannot be used for a normally clothed person at sedentary work where the metabolism is not basal, the various curves as shown in this paper would no doubt show a similar trend.

The conditions under which the observations were made cannot be considered as normal, inasmuch as the subject is prone with only about 60 per cent of the body

area exposed and the remaining 40 per cent insulated with a mattress. The effects of air currents on the body while in bed in a prone position will be much different than when standing or seated. The authors have recognized the fact that their method of skin temperature measurement does not give true skin temperatures. Perhaps a well-constructed, radiation thermopile, calibrated over the range of temperatures observed, could be used to better advantage.

No statement is made in the paper relative to the combinations of wet- and dry-bulb temperatures used to obtain the various effective temperatures. It would be interesting to have these same tests made for various effective temperatures using different combinations of wet- and dry-bulb temperatures to obtain the same effective temperatures. The correlation of skin temperatures might not be the same for similar effective temperatures obtained by different wet- and dry-bulb conditions.

One question I wish to ask before closing is: What is the physiological explanation of the fact that the skin temperature of the feet and hands were observed to be below room temperature in the cooler zone when body cooling by sweating is at a minimum?

PROF. C.-E. A. WINSLOW (WRITTEN): It is most gratifying to note the growing interest on the part of clinical investigators in the fundamental problems of thermal physiology, and this paper from the Department of Medicine of the University of Illinois is a valuable contribution to this subject.

The results in general tend to confirm and extend those of previous investigators in this field with interesting new data in regard to the vital capacity of the lungs. The increased vital capacity observed by the authors at high temperatures is to be expected, since at high temperatures the blood supply to internal organs, such as the lungs, should be reduced, and this reduction in blood supply should mechanically increase vital capacity. This point has not, however, been clearly demonstrated before and constitutes a valuable addition to our knowledge.

I note that the authors recognize the limitations of the method of determining skin temperature by the use of contact buttons. For exposed surfaces the type of thermopile, which measures surface temperatures by its radiation effect, is a far more reliable and satisfactory instrument.

DR. C. A. MILLS (WRITTEN): In discussing the paper by Hick, Keeton and Glickman may I point out that the first conclusion drawn from their studies needs some qualification. Their negative metabolism findings apply to body heat production only within the very limited period of exposure used by them (1-2 hours). The body mechanism for quick increase in heat produetion to avoid chilling is shivering or increased muscle activity. Changes in the *basal metabolism* come much more slowly, only after several days of adaptation to varying degrees of ease of body heat loss. These slower changes in internal heat production are very definite and have an important bearing on the conditioning of places of human habitation. The matter is one I have repeatedly stressed at successive Annual Meetings of the Society, but apparently I must still insist upon the point.

In the January bulletin section of *Heating, Piping and Air Conditioning*, E. Vernon Hill also takes occasion to emphasize his *belief* that the basic rate of body heat production plays little or no part in comfort sensations and in air conditioning problems. He states, "It (heat-generating in the body) is a constant quantity for each individual, depending upon the skin surface area. It does not vary at different times or at different seasons of the year, or even in changing air temperatures. The metabolic rate has no effect on comfort." Such statements, coming from authoritative sources within your Society and from those engaged in Society-sponsored research, may justly be expected to carry much weight with your members and others engaged in establishing a firm human background for air conditioning. But for that very

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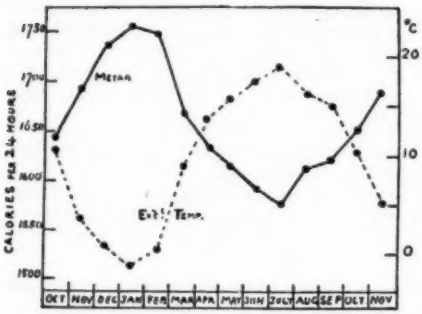

FIG. A. MEAN MONTHLY METABOLISM AND MEAN MONTHLY TEMPERATURE. GESSLER (1925), OBSERVATIONS ON HIMSELF †

same reason, they should be most carefully scrutinized for their accuracy of facts and deductions. In the present instance I fear their influence will be distinctly harmful to air conditioning. Hill's statements do not square with the findings of those who have looked for fluctuations in internal heat production under varying conditions of ease of heat loss. Nor have Hick, Keeton and Glickman in their reported studies provided any basis for conclusions regarding the effect of environmental temperatures on basal metabolism. Their period of exposure was only 1-2 hours, while several days are required to produce basal changes.

It might be well to show what changes in human basal metabolic rate have been found by those who have looked for them under changing conditions of body heat loss. Fig. A sets forth the changes in basal heat production as the body adapts to seasonal changes in temperature level through the year. In Fig. B is seen the marked suppression that took place during a sea voyage from Britain to Australia, through the moist heat of the tropics. I found a similar marked depression in basal heat production at the end of the hot 1934 summer in the Middle West. Others have made similar findings with men and animals,—a definite and marked drop in heat production

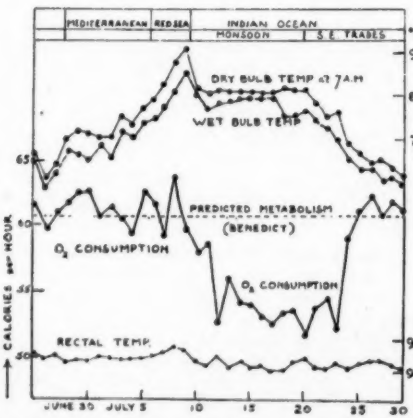

FIG. B. DAILY OBSERVATION OF BASAL METABOLISM OF C.J.M. DURING A VOYAGE FROM LONDON TO AUSTRALIA, JUNE-JULY 1923, AND DAILY RECORD OF THE TEMPERATURES OF THE DRY- AND WET-BULB THERMOMETERS AT 7 A.M.

† From Thermal Adjustments of Man and Animals to External Conditions, by C. J. Martin. (*Lancet*, 1930, 2, 617-620.)

as body heat loss becomes difficult. As body heat loss is facilitated by declining external temperatures, the basal heat production steadily rises.

Were we to deal, however, only with these differences in basal heat production, conditioning problems would be little affected, for people are at a basal level only when in bed with covers to prevent individual chilling. But the matter is far from ending there. Heat production during bodily activity is also proportional to that at bed rest. A person with low basal rate will spend less energy doing a given piece of work than will one with a higher rate of basal heat production. The tropical person, with his sluggish metabolism, is more *efficient* as a machine than is the energetic northerner. He will do more work on a loaf of bread as fuel, although he may take much longer to get it done. He wastes little in maintaining the high degree of muscle tone that is so common among more energetic people.

This difference in the metabolic cost of work, depending on the basal rate, is now becoming well recognized in medical scientific circles as a possible important factor behind the alarming rate of heart failure in the energizing regions of the earth, and particularly during the cold, stormy winter months when man is least efficient but most energetic. It plays a direct part, also, in the problems of properly controlling body heat loss by conditioning methods. Skin temperatures are bound to be much influenced by the total heat loss rate required by the body if all external factors be held constant. That fact is strikingly illustrated by one's comfort in a room while quietly at rest and contrasted with the discomfort, hot skin, and perspiration quickly experienced on vigorous exertion in the same atmosphere and physical surroundings.

We cannot, in our discussions of proper control of body heat loss, escape the necessity of considering these definite changes that take place in the heat production rate as environmental temperatures change or differ. Cooling of tropical mankind is quite a different matter (in degree at least) from the cooling of residents in the outer temperate regions. And, with severe summer heat, it must be remembered that even northerners soon drop close to tropical levels in their internal heat production and sensitivity to chilling.

Unfortunately for you engineers, it seems impossible to simplify heating and cooling standards, region by region, depending on average temperature conditions. Man himself is very responsive, within a few days' time, to changes in his environmental temperatures, varying his rate of internal heat production and hence his sensitivity to change in external conditions. A person with sluggish metabolism is markedly more sensitive to chilling, but less sensitive to heat, than is the more energetic individual.

More detailed and accurate data are needed regarding these normal fluctuations in different regions and through the seasons. It might be well for your Research Committee to undertake active support of such studies in order that air conditioning may progress on a firmer foundation. Please notice that I suggested *active support* of such work by the Research Committee, rather than attempts *at active direction*. Such studies must be carried out along medical and physiological lines, fields in which engineers (be they ever so expert) are not qualified to direct.

W. L. FLEISHER (WRITTEN): The continuation of research on the Physiological Response of Man to Environmental Temperature, as covered by the paper by Doctors Hick and Keeton of the University of Illinois Medical School and Nathaniel Glickman, our Research Fellow, I feel, has touched on one point which is of great interest to society only indirectly, and, like most of the research of this kind, does not indicate the effect of evaporative temperatures as affecting the physiological response of man. In other words, the thermopile method of taking skin temperatures precludes the effect of evaporation, due to the fact that the thermopile must necessarily

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be held to the skin by tape or plaster, which makes the thermopile impervious to an evaporative response.

The paper would indicate that the basal metabolism of the subjects varied almost as much at 70 deg ET as it did at 80 deg ET, but it would be interesting to know exactly what dry-bulb and wet-bulb, and what air movement created the effective temperatures utilized in the experiments. For instance, in Fig. 3 the variation in basal metabolism is from 37 cal (calories) per square meter per hour to 45 cal, a difference of approximately 33 per cent. This variation persists throughout most of the experiments on most of the subjects.

On page 150 the authors state: "Basal metabolism is not a function of the wet-bulb", and still they contend that the effective temperature is a better coordinate of physiological reaction than either the dry-bulb or the wet-bulb. According to my interpretation they seem to contradict this on page 155, where they state: "Fig. 6 indicates the less regular distribution of points when skin temperature is plotted against dry-bulb". They continue: "It is well known that the amount of heat lost from the body by radiation depends on the temperature and character of the walls; that the heat lost by evaporation depends on the wet-bulb, the dry-bulb, air velocity and type of air movement; and the heat lost by conduction and convection on dry-bulb and air motion". Until we understand at exactly what dry-bulb and wet-bulb the effective temperatures were plotted, it will be impossible to comment analytically on this particular point.

The authors' comments on the onset of sweating are interesting but, in my opinion, incomplete. In the recommendations for summer comfort cooling, the disappearance of the visible moisture on the forehead indicates, usually, the comfort equilibrium point. This undoubtedly is a function of both the dry-bulb and wet-bulb and air movement, and our records show that a condition of 81 deg ET which according to the authors was about the point where sweating began, is far above any effective temperature that we recommended.

I believe that the investigation that has been carried on is preliminary to some extremely interesting developments and that, in general, it confirms the results that Dr. Sheard of Mayo Foundation presented to us in his paper [27] of last year. However, I feel that this work must be carried on further in order to be helpful to either the medical or engineering profession.

I might take exception to some of the conclusions. When a statement is made, that the chemical regulation of body temperature is negligible under the conditions investigated, I would say that I do not believe that there are enough facts included in this paper on which to base such a conclusion. If chemistry and electricity are as closely related as they appear to be, then electrical tests should be associated with the tests that were made before such a conclusion can be arrived at. The Medical School of Yale University has been making electric nerve emanation tests, which indicate wide variations in electrical output from various nerve and surface centers, which would indicate, possibly, contrary conclusions.

In the authors' conclusions they also state: "Radiation of heat is a principal means of heat loss in a range below 80 deg ET. It is controlled by a sensitive regulation of skin temperature in the extremities." I feel sure that wet-bulb temperatures must enter into this regulation, because other investigators have discovered that the extremities are often below the dry-bulb temperature of the enclosure itself.

In connection with Conclusion No. 4 the investigation under other research projects has indicated that subjects entering a conditioned enclosure after a slight amount of exercise had rectal temperatures above the normal or so-called normal, and this was

[27] Investigations on the Exchanges of Energy Between the Body and Its Environment (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 115).

after coming from environments which certainly were not above effective temperatures of 91 deg. There are, therefore, other factors in normal human beings which would tend to raise the rectal temperature and which, in a way, are not indicated in this paper.

Under Conclusion No. 7, the criticism that I have made before still holds—that without a knowledge of the actual dry-bulb and wet-bulb temperature under which these tests were made, it is difficult to analyze either the results or the conclusions. The variation in the points charted in the various figures is a much wider variation than ordinarily would be tolerated in scientific investigations, and possibly between the variable points lies all of the solution to the problem at issue.

In my conclusion I want to say, that I feel the paper indicates certain interesting and fundamental results which should be broadened and clarified in order to utilize the experimental work that has been done.

PROF. C. P. YAGLOU (WRITTEN): Although this paper presents little that is new, it is a valuable contribution to the literature on heat regulation in that it clarifies and emphasizes the important factors concerned.

Accepting the fundamental fact that, in persons at rest under ordinary indoor conditions, metabolism and deep tissue temperature remain substantially constant, winter or summer, the problem of evaluating conditions of comfort resolves itself to a study of factors affecting quality, as well as quantity, of heat loss. The problem appears to be much more complicated than was thought to be a few years ago, when the degree of warmth and comfort were being evaluated on the bases of such simple criteria as total heat loss, and forehead or cheek temperatures. Numerous recent studies, supplemented by the observations of the authors of this paper, show clearly that the extremities, particularly the hands and feet, are the principal heat regulators of the human body, and as such deserve much more attention and study than heretofore afforded. There is also much good evidence to show that seasonal adaptation is largely effected by distinct changes in the temperature of these surfaces, with comparatively minor changes in other parts of the body. This holds true until the onset of sweat when the skin temperature becomes uniform all over, as shown clearly in the authors' charts.

A minor point of disagreement is in the authors' reference to man's inability of lowering heat production in warm atmospheres. Although the literature is contradictory, there is ample evidence to show that prolonged exposure to tropical heat does lower the basal metabolic rate (Sundstroem, Mason, Martin, de Almeida, Mills, Montoro, and others).

DR. E. V. HILL: We have some very pertinent and important discussions of this paper, so I will take a very short part of your time. I just want to mention one or two things that I think should be brought out at this time. The first one is that the appropriation for carrying on this work was made by our Society, but there was no appropriation made to install or provide the equipment in the experimental rooms, and it is due to the efforts of John Howatt that we were able to get the apparatus necessary for air conditioning and controlling the experimental rooms.

I will not tire you with a lot of detail as to the trouble and difficulties you may get into if you try to take apparatus from a dozen sources and correlate it in an installation of this kind. But I do want to say that due to the work of our committee the room now is in ideal condition. We have a perforated ceiling that gives us a distribution of air in the room that is as near perfect as anything I have ever seen without any objectionable draft. We have good equipment and the staff are doing a splendid job in getting started in this research program. The paper you have heard, excellent as it is, is only a foreshadowing of what we will expect from Dr. Hick, Dr. Keeton and their associates in the next year.

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There are only three points I want to mention in the paper. One is the subject Dr. Hick mentioned of vital capacity, and I think that we may expect some interesting development along this line. The vital capacity of the lungs increases at the higher temperatures which, to me means that the blood cannot be in two places at once. It is in the skin surfaces to aid in heat dissipation, therefore it is not in the lungs.

Dr. Arnold brought out this same point in a paper before the Illinois Chapter recently, arguing that this same condition reduced the blood supply to the digestive organs, and he thinks it accounts for a considerable amount of digestive disturbances, particularly in children. It is a line of investigation I think well worth following.

One other point which Dr. Hick has brought out and emphasized is the importance of the hands and feet as regulator organs in heat dissipation. As you may recall, this was first brought to our attention forcibly last year by Dr. Charles Sheard at the St. Louis meeting. Dr. Hick has amplified and emphasized this very important thing. We now have a different picture of how the body adjusts itself to changes in environmental conditions, and the great importance of the hands and feet. Most certainly we must go over very carefully the work previously done on effective temperature and, I think, correct the data.

The last point I will make is that these tests in connection with research studies at Illinois are being done with nude subjects, or practically nude subjects. I have always been able to get a laugh for many years when I insisted that we should work with nude subjects in determining something basic with regard to the effects of the air environment on the human body, and it is a pleasure to me to note that Dr. Hick and his associates and also Professor Winslow are now coming to my support with regard to this subject of nudity.

PROFESSOR WINSLOW: I was particularly glad to hear Dr. Hick's emphasis on the importance of measuring all the factors concerned in the atmospheric environment. The effective temperature was a great step forward, but we still need to take account of radiation effects.

At the meeting of the League of Nations Committee on the Hygiene of Housing last June, it was recommended that all future studies in the field of air conditions in houses (and the same should apply elsewhere) should include four independent measurements of the four independent variables concerned. When these four variables are known they can be combined into any sort of effective temperature or operative temperature index that one desires. As long as the air temperature, the mean radiant temperature of the surrounding surfaces, the relative humidity and the air movement are not separately measured, it is impossible to compare the various indices.

The Committee on Hygiene of Housing recommended that these measurements should be taken with the following four instruments, or with equivalent instruments: dry-bulb temperature by an ordinary thermometer shielded against radiation effects; the mean effective radiant temperatures of the surroundings by the Vernon globe thermometer, its readings, of course, taken in conjunction with those of the other instruments; the relative humidity by the ordinary psychrometer, and the air movement either by the silvered Kata thermometer or by the hot-wire anemometer.

I hope very much that in the future we may have published data which will give us these four independent factors, the only four factors that are concerned, and then and only then can we form a really complete idea of the thermal effect of the environment.

DR. MILLS: It is particularly gratifying to see the care with which the immediate body reaction to environment is being studied. The report of last winter at your January meeting by Sheard and Williams and this year by the authors of this paper is giving some very good evidence on how the body reacts acutely over a short period of time to the environment. That reaction seems very definite and it involves par-

ticularly wide changes in the circulation and surface temperatures in the hands and feet. We must bear in mind, however, that immediate reaction *is* an immediate reaction and that it is rather strongly conditioned on slower reactions in the body that are determined by the basal heat production level.

Dr. Hick in presenting the paper made a qualification that made rather unnecessary the remarks I had prepared, because he stressed the fact that their studies were made only over short periods of time, which means of course that the basal metabolic readings, or the rate of heat production determinations which they gave, have very little bearing. Their first conclusion bore on this subject, stressing the fact that they had negative metabolism findings applying to body heat production, but this was with very limited periods of exposure, one to two hours. The body mechanism for a quick increase to avoid chilling is shivering or increased muscular activity.

I should like to support the statement of Mr. Adler that there is seriously needed by your Society a careful statement of objectives. You do need much more information on what man does when faced with certain environments, and what your changes in environment are going to produce. I believe your Society should broaden its active sponsorship of such types of work. Notice that I say *sponsorship*. I certainly feel that the *direction* of that type of work should fall into medical hands. Be you engineers ever so expert, you are not qualified to direct physiological work; but that is the type of work which needs to be done before air conditioning or any type of conditioning for our own welfare will really be on a firm basis.

N. D. ADAMS: I want to take this opportunity to congratulate the Society on this cooperative research; more especially I want to take this opportunity to extend to the Society the greetings of Dr. Sheard of Rochester. He is very interested in this paper, in that quite a bit of the work correlated with the work that he presented last year. During this last year he has been working on problems where the patients are affected with various circulatory diseases, loss in circulatory control, and he has found some very interesting things.

Further, I think it is important that this question of humidity and dry-bulb should be studied further. We tried to correlate some of the points on this and found that we could not correlate the points where the metabolism was correlated with the dry-bulb and with the effective temperature. I think the other point I wished to discuss has been mentioned.

DR. F. K. HICK: I wish to thank the several discussants for their very friendly and interesting discussion. Dr. Hill has helped us rather consistently with certain of the numerous technical problems involved in this work.

In answer to Mr. Aeberly, we have no data on the carbon dioxide content of either the arterial blood or the alveoli in these patients. We are equipped to carry out that sort of procedure and have done so many times on other subjects. It is something of a job. I would think offhand, from indirect evidence, that CO_2 effects are negligible in this whole adjustment process chiefly because the ventilation of the lungs per unit time does not increase until the heat becomes intolerable.

In answer to Mr. Rodee's discussion, I am familiar with the radiation type of thermopile and appreciate that it has many advantages over the ordinary button type. We were able to calibrate our button type instruments with the aid of A. E. Hershey, assistant research professor of mechanical engineering, University of Illinois, who found in a series of tests conducted over a range from 79 to 106 deg dry-bulb that the temperatures recorded on the surface were within 1 deg when given by our button type thermopile, with those figures further determined by radiation thermopile. One reason for not using a radiation thermopile is that we found a type K potentiometer in the surplus equipment room and constructed the balance of the equipment that we used for this thermopile at a total cost of less than $6.00.

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The physiologic principle by which the hands and feet come to be at a temperature lower than room temperature, measured by dry-bulb thermometer, is most interesting. As has been said, many other people commented upon the fact that the hands and feet can actually get colder than a chunk of wood placed in the same position could get. Presumably that is because of the surface acting as a wet surface and refrigerating action taking place in that manner.

Professor Winslow mentioned the importance of radiation measurements in defining what an environmental condition is, and if I brought anything new to this Society it is an emphasis on that necessity.

Dr. Mills' discussion is most interesting. I am familiar with the extensive literature involved on the changes of basal metabolism and acclimatization to warm environments. I wish to comment only that that subject is by no means unanimously decided one way or another, and that there are controversial points in that affair. I think our data confirm those presented by DuBois in his lecture at Stanford University, recently published, and the work of Winslow and his associates in failing to find such variations in anything like acute circumstances. Our control over these patients has exceeded one or two hours, and has ordinarily been from 12 to 14 hours.

Mr. Fleisher's discussion has many points of much interest in it. His emphasis on the importance of a wet-bulb measurement as some function of physiologic processes I cannot contradict. I do not have enough experience to do so. We plotted these points against effective temperature scale perhaps without sufficient reason, and I would be very happy, if your Publication Committee is willing, to add a supplement to this paper giving in much more detail the wet- and dry-bulb temperature involved.

The ability to study electrical processes and to associate them with oxidative processes in the human body is probably beyond the experience of our group. I have had no experience with such things and am unable to comment on the studies carried on at Yale University.

Mr. Fleisher comments on the scatter of points as being almost beyond acceptability for scientific investigation. I agree with him heartily. In fact, for many weeks at the start of this investigation, we were almost ready to abandon certain parts of it because of an inability to get consistently reproducable results, but that is not chance, however, I will insist. This scatter represents something real, but what it is I do not know. That is part of it. The other part of it is that in biological processes there is always a scatter which is not a part of physical processes. Engineers are accustomed generally to working with concise data many times accurate to five and six and eight significant figures. Biological data that have more than two significant figures are almost certainly erroneous and artificially accurate. Illustrations show the rectal temperature of certain patients, which again shows a variability much higher than might be expected, because at a given temperature, for example 80 deg, on this subject, that man's temperature varied (each of these points represents a different day) as much as 1½ F.

Here again the variation in this supposedly constant situation is again practically 1½ F, with the temperature condition one day and another identical, 68 deg ET.

There again another patient, not quite so stable perhaps, shows rather consistently a variation of 1½ F at the same conditions from one day to another.

That is a plea for tolerance of biological data, of the type of error which is not inherent in physical measurements themselves.

Professor Yaglou's discussion again brings up the item of seasonal adaptation.

Mr. Adams presents Dr. Sheard's greetings. I had the pleasure of a very pleasant day with him last month and was very much interested in hearing that many of his difficulties were those that we had experienced.

No. 1080

CONTROL OF BODY HEAT LOSS THROUGH RADIANT MEANS

By C. A. Mills, M.D., Ph.D.* (*MEMBER*), and Cordelia Ogle, Ph.D.** (*NON-MEMBER*), Cincinnati, Ohio

IN this article it is desired to present the results of certain preliminary studies which strongly indicate the feasibility of controlling body heat loss through radiant heat channels. These results appear to be striking and unequivocal in their significance, so much so that their early publication was deemed advisable in order that other investigators might aid in expanding this line of research. Briefly stated, efforts have been made to devise a conditioning system whereby body heat loss will be properly controlled, either in summer heat or winter cold, through radiant heat input or removal, quite independent of air humidity or air and wall temperatures.

Present *air* conditioning methods possess certain definitely unsatisfactory phases which are obviated in the new set-up. First and most important is the present necessity of maintaining sharp contrasts between indoor and outdoor air temperatures, both for winter heating and summer cooling. Respiratory infections, which form the great bulk of human illness in stormy regions, seem definitely related to storm changes in the outdoor atmosphere, particularly to sudden changes in temperature. There is much ground, therefore, for suspicion that such infections are also promoted by the necessity of facing sharp contrasts in air breathed on passing from a 75 F room into outdoor air at 0 F, or from outdoor air at 100 F into a room cooled from 80 to 85 F. The lesser contrast faced in summer heat may be just as effective in promoting respiratory infection, because of the lowered body resistance that is customary during hot weather.

The second objection to present methods lies in the necessity of warming or cooling the whole air mass and wall materials in order to produce body comfort. With air seepage and conduction loss from buildings, and with the human needs actually forming only a small part of the heating or cooling load, air conditioning would seem to be highly inefficient as a method of controlling body heat loss. More direct transmission of heat to or from the skin and clothing, without regard to air conditions, is possible only by radiant means. It therefore becomes important to know whether it is physiologically sound to control body heat loss in this manner.

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DESCRIPTION OF EXPERIMENTAL ROOMS

Experimental rooms were set up within larger laboratory rooms and equipped to cover the desired conditions. In Room 1 (6 ft x 6 ft x 6½ ft inside measurements) with inside walls and ceiling constructed with rigid insulating board, air temperatures were maintained at 89 to 92 F and relative humidity at 60 to 70 per cent. Room 2 was similarly constructed, with air temperatures and humidity kept at the same level as in Room 1. However, it was lined throughout with aluminum foil glued to the wall board and contained six cold plates (20 in. x 42 in.) which were installed on two opposite side walls and connected to a condensing unit located outside the enclosure. These plates were kept at 32 to 40 F and acted as absorbing surfaces for radiant heat liberated within the room. They were exposed to the air and naturally imposed a heavy load on the air conditioning equipment by their cooling and dehumidifying action. In both rooms, however, air conditions were satisfactorily maintained at the desired temperature and humidity levels, and with not over 2 F difference in air temperature at points one foot from the ceiling and one foot from the floor. Room 3 (8 ft x 8 ft x 7 ft double walled steel, with 1 in. rock wool insulation) was also lined with aluminum foil and equipped with an air cooling unit connected to an outside compressor. In addition it had arrangements for an input of radiant heat from ordinary coil reflector units (400 watt total capacity). With no heaters operating, the air temperature could be reduced to 32 F, and with all heating and cooling capacity in use, air temperatures stabilized from 70 to 76 F, varying slightly with outside temperatures. Room 4 (similar in construction to Room 3) was air conditioned from a 66 to 70 F level by an ordinary room conditioning unit, with no radiant heat or cooling surface exposed within the room. In all four rooms adequate and similar ventilation was supplied by exhaust fans and proper air inlets, using inside laboratory air. Air conditioning within the rooms was accomplished by recirculating units placed inside the respective rooms.

ENVIRONMENTAL TEST RESULTS

Effect on Human Subjects. Preliminary tests on human subjects indicated distinct discomfort in Room 1, with free perspiration and a smothering sensation. In Room 2, with identical air conditions, the stuffiness and smothering sensation were lacking, and adequate body cooling took place. In fact for a person to sit there working several hours, it became necessary to cover over part of the cold plates to prevent actual body chilling. One of the subjects, in caring for the mice during that part of the study, found it necessary to wear a woolen sweater in this room, even at 89 to 92 F air temperature and 60 to 70 per cent relative humidity. In Room 3, with the air at 32 F and no radiant heat on, body chilling was severe; but delightful comfort was achieved within a few minutes after the heaters were turned on, even though air temperatures were still below 40 F and the exhaled breath still visible. After all heater and cooler units had been in operation for days or weeks, this room was uncomfortably warm for human occupants within a half hour's time, at air temperatures of 70 to 76 F, no matter how lightly clothed. In Room 4, subjects were cool but comfortable when dressed in usual indoor winter clothing.

Effect on Mice. Human comfort for brief periods, however, is not a safe

measure of environment effect. More general biologic effects after prolonged exposure must also be studied and taken into account. Accordingly, groups of healthy young white mice, which had been grown in the control room, were placed into the four test rooms. Every precaution was taken to have the four groups as similar as possible when placed in the rooms at three weeks of age. All were fed the same optimal diet which was used in previous work.[1] Weekly weighings were made on each individual mouse to get the growth curve, and the age at onset of fertility was obtained by frequent matings with adults of known fertility as soon as the mating age was reached. Each group consisted of 10 males and 20 females.

In Table 1 are given the results of these mouse observations. It is seen that fastest growth and earliest sexual development occurred in the group kept

TABLE 1—GROWTH AND DEVELOPMENT OF WHITE MICE UNDER VARIATIONS IN EASE OF BODY HEAT LOSS

	ROOM 1 MOIST HEAT	ROOM 2 MOIST HEAT-RADIANT COOLING	ROOM 3 COLD AIR-RADIANT HEATING	ROOM 4 AIR COOLED
Average weight in grams of males when placed in rooms at 3 weeks of age....	11.1 grams	10.8 grams	10.8 grams	10.5 grams
Same at 10 weeks of age.....	26.1 grams	29.3 grams	27.4 grams	28.3 grams
Same at 15 weeks of age.....	28.3 grams	32.0 grams	29.4 grams	30.2 grams
Average age in days of females at delivery of first litter.....	63.9[a] days	55.1 days	56.3 days	56.2 days
Average number per litter ...	6.5	7.7	7.6	7.7

[a] Half of the 20 females in this room have still shown no fertility at 16 weeks of age, even with repeated matings.

in Room 2; these animals developing even more rapidly than did those in Room 4 where best development had always previously prevailed. In Room 1 growth and development were suppressed by the moist heat, similarly as had been found in previous studies. In Room 3 a suppression of growth also took place, even though air temperatures were only 70 to 76 F and relative humidity low. Good health was maintained by the animals of all groups, with not a single death during the four months of observation. Accompanying the growth stimulation in Room 2 there was an earlier onset of fertility and the birth of litters about 1/5 larger in numbers than in Room 1 where growth suppression was accompanied by the birth of smaller litters of less viable young.

Whether judged by human comfort or mouse growth standards, it is evi-

[1] Adaptation of Sexual Activity to Environmental Stimulation and Climatic Influence on the Growth of the Male Albino Mouse, by Cordelia Ogle (*American Journal of Physiology*, 1934, Vol. 107, pp. 628-640).

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dent that in Room 2 the depressing effects of moist warmth have been completely obviated by facilitating loss of body heat by radiation. With air at 89 to 92 F (practically summer dry skin temperature) and relative humidity of around 65 per cent, heat loss by conduction and convection can be only slight, and only by free perspiration and rapid air movement can evaporation be effective. The finding that body comfort and optimal animal growth can be secured in such a room, below the perspiration point, means that removal of practically all body heat *through radiant channels* is physiologically possible, and apparently quite desirable. The metabolic depression of Room 1, quite typical of tropical moist heat conditions, can be completely overcome by facilitating radiant heat loss alone, without any change in air temperatures or humidity. Just why the air of Room 2 should be so much more pleasant to breathe than that of Room 1 is not at present explainable, since they are apparently identical. The effective factor may perhaps concern radiant heat loss from the nasal mucosa directly.

The suppression of growth of the mice in Room 3 illustrates well the contention that it is the ease or difficulty of body heat loss which determines the metabolic or internal combustion level, and all the other vital functions thereto attached. No matter what the air temperature level, if the radiant heat load falling on the skin is so great as to make difficult the dissipation of this heat and that internally produced, then the body's response will be a reduction in internal combustion and a slowing down of its metabolism. Had facilities been available, it would have been desirable to run a fifth group of animals in Room 3 at air temperatures around 40 F, to show that optimal growth could have been safely produced by the right amount of radiant heat input, even though mice are very susceptible to body chilling. It is hoped that a series of tests can be run and the results reported soon.

Problems in Heat Physics

Certain interesting problems in heat physics have arisen during this work. Lack of suitable apparatus has prevented anything more than a cursory investigation of these physical aspects of the study. Unfortunately physical research seems never yet to have dealt with a number of radiant heat relationships that are here of fundamental importance. Aluminum foil, for instance, is known to be a poor emitter of heat and of high reflectivity; but does it reflect heat of all wave lengths equally well? For winter heating by radiant means, shorter wave length radiations from a hot surface are of concern, but in summer cooling the wave lengths dealt with are much greater. Nor is there available information relative to the absorption or emission of radiant heat from the different types of materials used in clothing and room furnishings.

Air and Wall Surface Temperatures

A few crude observations of air and surface temperatures were made in the test rooms, with results as indicated in Table 2. For air temperature determinations, ordinary mercury thermometers were used, after insulating against radiant heat by foil shields. Surface temperature measurements were made by a distant reading resistance thermometer.

The data of Table 2 afford an insight into the mechanism of radiant heat

transfer that renders this method of controlling body heat so effective. In Room 1 with no foil, all materials and the air assume a uniform temperature, making body heat loss by all avenues difficult. In Room 2, air and foil wall surfaces are at the same temperature as in Room 1, but all other materials assume definitely lower temperature levels, due to loss of heat by radiation to the cold plates. The thermometer hanging in this room exhibits a similar drop in temperature if its foil shield is removed so as to expose it to heat loss by radiation.

In Room 3, on the other hand, it is seen that air and foil wall surfaces tend to assume the same temperature soon after the radiant heaters are turned on, and that heat absorbing materials in the room (wood, iron, cloth, etc.) tend to have temperatures progressively higher than air or wall surface levels. And in this warming up process of heat absorbing materials, the most rapid temperature rise takes place in cloth. Preliminary observation indicates also that fall in temperature in the foil-lined hot room takes place more

TABLE 2—AIR AND SURFACE TEMPERATURES WITHIN ROOMS

MATERIAL	TEMPERATURES, DEG F					
	HOT ROOM 1 (NO FOIL)	HOT ROOM 2 (FOIL-LINED)	COLD ROOM (FOIL-LINED) RADIANT HEATERS ON OR OFF			
			Off for 3 Hours	On for 10 Min.	On for 40 Min.	On for 18 Hours
Air	90	90	50.0	58	66	76
Wall Surface	90	90	54.5	61	66	76
Wood Block	90	83	56.5	67	74	85
Iron Block	90	83	56.5	64	73	85
Cloth	90	83	...	77	80	85
Trousers (on wearer)	91	85	...	82	85	89

rapidly from cloth than from wood or iron surfaces. No detailed studies have yet been made as to differences in heat absorption or heat loss by different kinds of cloth. It will be of interest to see whether woolen and linen materials differ in this respect.

GENERAL CONCLUSIONS

In summary these preliminary findings indicate certain important points: (1) Human comfort and animal growth and development can be regulated, with what seems apparent safety, by control of body heat loss through radiant channels alone; (2) Practical feasibility of such radiant control lies in differences in radiant heat absorption and emission exhibited by various substances. Clothing materials rapidly radiate their heat to cold surfaces (if such be available) and fall to temperature levels well below those of surrounding hot air, thus encasing the body in a cooled capsule to which it can readily lose its internal heat. And in cold air, clothing readily absorbs radiant heat and forms a warm envelope around the body to slow the process of heat loss from the skin; (3) Foil wall surfaces act as efficient passive reflectors of radiant heat, although their temperatures are always maintained at air temperature level. This passive behavior of foil surfaces would seem largely

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to obviate the need for wall insulation, since the temperature of the wall surface is no longer of importance.

Certain problems in engineering and physics remain for solution. Foremost of these is the necessity of foil decoration that will not impair its heat-reflective property. Present physical tables of heat transparency for pigment materials which might be used in such decoration do not cover the longer wave lengths such as are radiated from skin and clothing. It is known that certain lacquers may be applied without much alteration in the foil reflection of radiant heat, but even in this case there is no certainty that the lacquer might not absorb the longer skin radiations. A second problem, in engineering, lies in the development of means to isolate the cold plate from warm room air and moisture, so that air cooling and dehumidification may be avoided. Few known materials are transparent to heat waves from the skin, rock salt crystals being the only common material of high transparency.

Success in the preliminary studies here reported should go far to stimulate thorough investigation of the possibilities of adapting this scheme to the conditioning of indoor environments. It seems physiologically sound and to offer certain important advantages: (1) It makes possible elimination of marked contrasts between indoor and outside air; (2) The heating or cooling load will of necessity be far less than that of present conditioning methods, since the air and building mass no longer need so much warming or cooling; (3) Change to electric energy sources, reduction in insulation needs, and possible changes in type of construction—these need not be more than mentioned at this time.

Closest possible cooperation between engineers and physicists on the one hand, and medical and physiological scientists on the other, will be essential to the proper investigation of the possibilities residing in the conditioning scheme dealt with in this presentation of preliminary findings.

DISCUSSION

T. N. ADLAM (WRITTEN): This article is very timely and presents much valuable information, and much credit is due to the authors for the trouble they have taken to obtain information on such an important subject. It has for years been obvious to some of us that since we have not been able with certainty to lay down what physical conditions will give comfort with health, the whole scientific basis of heat engineering has been in doubt.

The information given in the article substantiates something we have known for years, viz., that the human body, while not requiring any heat from without, can, within limits, be kept from the sensation of feeling cold in two ways: first, by raising the temperature of the air surrounding the body to a sufficiently high point; and second, by allowing thermal radiations from a warm object to impinge on the body with sufficient intensity to make up for a lower air temperature. The converse being equally proved in keeping the body cool.

The generation and losing of heat by the human body is a complicated process because the feeling of hot or cold does not appear to be a measure of heat loss to heat generated inside, but rather that the sensation of the body is more susceptible to the way in which heat is abstracted from the body.

I think the authors are perfectly correct in stating that present air conditioning

methods possess certain definitely unsatisfactory phases which can be obviated in the new set-up. Undoubtedly, some conditions stimulate the production of heat within us which enables the body to respond to the environment and generate more heat to meet the conditions; on the other hand, some conditions increase the dissipation of heat without in any way controlling the generation of heat within.

I think the principal reason why present air conditioning is unsatisfactory is that the factor for radiation has been neglected, and I believe no system will be 100 per cent satisfactory unless the three principal factors, viz., radiation, convection, and evaporation, are taken care of in their correct proportion as demanded by nature.

I agree absolutely with the authors that the large temperature differential which heating engineers have advocated in the past, both for summer and winter, is not in accordance with true comfort and health. I think too much emphasis has been given to the condition of the air and the easiest method to attain that condition instead of studying the true physiological demands of the human body.

We know that the human body is readily adaptable to varying conditions over a small range and, if the emergency mechanism such as sweating is considered, over a wide range.

However, it should be the object of a heating or cooling installation to avoid the necessity for adaptation, although the air may be made chemically pure, a big temperature change offers greater risk to the human body than a system with a smaller temperature differential taking care of radiation and humidity with natural fresh air even though not chemically pure. It seems fairly obvious that the reason the air in Room No. 2 felt more pleasant than the air in Room No. 1, though at the same temperature, was that the human body could give off its surplus heat in the former by radiation, thus allowing the organs of the body to function freely and burn up any surplus carbon. If the human body cannot dissipate its surplus heat freely it will crowd up on the surface of the body and cause discomfort and lassitude.

The authors, on referring to wave lengths, did not give us the surface temperature of the radiant units used; whether they were hot enough to be visible or whether they were of low temperature type. It has been found in England that low temperature radiation gives the best results as there is less convection current created and no risk of burning the air, as is possible with high temperature radiant units where particles of dust are split up chemically and ammoniacal vapors given off.

What we call heating by low temperature radiation is not really heating at all, except in a secondary sense. We produce low temperature radiation not in order to heat the man in the room, but to reduce the net rate at which the surface of his body parts with his own heat by radiation according to the physiological demands of the body.

I notice ordinary mercury thermometers were used to take the air temperature and resistance thermometer to measure the surface temperature, so that I imagine the results obtained by the investigators in this way were only approximate because as soon as an object, however small, is placed over a surface which is normally receiving or giving off radiations, the conditions are at once altered. I think readings taken by a thermopile would have disclosed different wall surface temperatures when using radiant heat to those when using conditioned air only. Much information is needed to tell us the best surface temperature to be used and comparisons of the effects of different orientation of the heating surfaces.

As to the wave lengths reflected from the aluminum foil wall surface, it would be very interesting to know what wave lengths are reflected from all surfaces usually used for interior decoration. However, we naturally obtain the reflected waves in addition to the longer waves emitted from the surface as it becomes gradually heated by the absorption of heat rays from the radiant elements.

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Usually, we find that with low temperature radiant heating the wall surfaces are three or four degrees above the air temperature, and several degrees lower than the air temperature when convected heat only is used. These surface temperatures will, of course, depend very largely on the actual surface, the construction of the walls, and the surrounding temperature conditions outside the room.

If, as one must assume, the experimental rooms were set up inside an already heated larger room, the conditions are vastly different to the conditions which would prevail if the rooms were exposed to extreme outside weather conditions. Undoubtedly, the effect on the human body can be well studied under certain predetermined conditions, but it is a commonplace that in any miscellaneous crowd we meet wide differences of opinion on the question of comfort; a word we often use but a word full of ambiguity unless it is comfort with physiological compatibility, since the conditions of comfort are not necessarily synonymous with conditions of health. People living under conditions entirely opposed to physiological compatibility will often confuse a feeling of warmth with that of physiological comfort. The authors have done good work in proving once again that the human body can obtain more comfort when subject to the effect of radiation instead of being stifled in winter with air too warm, or chilled in summer with air too cold.

The method of dealing with our requirements by radiation is closely allied to nature's way, for the sun sends us short rays which can penetrate the earth's atmosphere and which, impinging on the earth's surface and surrounding objects, warms them up so that they send back to us longer heat waves to protect our bodies from radiating too much heat to outer space.

The authors, in their general conclusion, draw special attention to the indication that human comfort and animal growth and development can be regulated with what seems apparent safety by controlling the body heat loss through radiant channels. This has been well substantiated in the radiant heating systems already in use where a true balanced account is kept on the body temperature.

Much investigation is needed, however, because with the present materials at our disposal it seems impractical to maintain high surface temperatures in winter without quite appreciably increasing the air temperature, and in summer it will be equally difficult to maintain surfaces at such a very low temperature as given without gradually lowering the air temperature and depositing moisture on the surfaces.

It is claimed by the authors that the temperature of the wall surfaces is no longer of importance. I think that when these tests are extended to a more practical conclusion it will be found that wall construction and the surface temperature will be important factors in both comfort and fuel consumption. I think also that the future will find heating engineers specifying a well constructed building with well lagged walls and ceilings, radiant surfaces for either heat or cold protection to give comfort with cooler air temperature in winter and comfort with tempered air in summer, with plenty of unadulterated fresh air, with humidifying or dehumidifying for winter and summer respectively.

While the tests made by the authors have proved that comfort can be maintained by the introduction of radiation either positive or negative, it would not be wise to draw too many definite conclusions from these preliminary tests, otherwise we may be guided along another false avenue as we have for years been traveling along the wrong avenue as regards air temperature and air movement necessary for comfort.

We are, undoubtedly, up against an elusive problem which requires very careful thought and investigation and it must be realized that any data gathered from small rooms constructed in a laboratory can be only tentative. Ultimately, the standard coefficient figures and conditions on which actual calculations will be based will be derived from actual installations. We have already a sufficient number of radiant

heat systems in this country to know that it is suitable to our climate and gives excellent results with lower air temperature and less fuel consumption.

The conditions in such small rooms as are usually used for investigations are so different from those of practical life that I have doubt as to how far any results that can be obtained would be applicable to practice. This was my experience in the early days of radiant heating in England, and while the experimental data obtained are invaluable as a guide, we have to be careful not to make definite decisions until we go out into the wider field of practical experience.

We owe very much to the authors for the trouble they have taken to obtain these data and for giving the results to the advancement of heat engineering. I, for one, sincerely hope it will be given all the consideration it deserves because I believe it is advancing in the right and only safe direction.

PROF. C.-E. A. WINSLOW: I think it is extremely interesting, this *rediscovery* of radiation influences. I say *rediscovery* because Pettenkofer, the great German hygienist, of the seventies, illustrated the phenomena by describing the sensations of the traveler on entering an inn where the fire has just been lighted and the walls are cold. Then it was forgotten for 30 or 40 years; everybody ignored the fact of radiation influences. Recently, of course, there has been a mass of work done upon it. Dr. DuBois and Dr. Hardy here in New York, and our group in New Haven, have been busy with this problem.

Dr. Hick presented an extremely important contribution and the problems are beginning to be understood. Some of Dr. Mills' doubts as to our knowledge of certain fundamental physical data are not quite warranted. I think if he will talk to Dr. Hardy, the physicist engaged in the Cornell work, and Dr. Gagge, our physicist, he will find that all the needed data as to the radiant behavior of various metals at various wave lengths can be easily located in the literature.

In England, of course, this field of radiant heating has come into practice. It is one of the most important standard methods now in use in England and has proved extremely successful. I am afraid it will not, as Dr. Mills suggests, do away with the necessity for convection insulation, by substitution of radiation insulation. Even if we had our inside walls covered with aluminum, it is obvious, I think, that the radiant heat supplied will all ultimately be converted into convected heat. I believe in radiation insulation, but I think that behind your aluminum you should have convection insulation to complete the job.

What Dr. Mills has been dealing with here are the effects of what we have called operative temperature. What he has shown is that you can neutralize the effect of a hot atmosphere by cold walls, and what we have called the operative temperature is a measure of the combined effect of wall and air temperature.

The problem in which we are all particularly interested is whether, with the same operative temperature, that is with an identical thermal influence, there are any specific advantages or disadvantages of radiant as compared with convected heat. That question is not answered yet by Dr. Mills' work. He has shown that animals in a room that is too hot for them can be relieved by radiation to cold walls. What we really need to know, and the problem with which we are struggling and which perhaps Dr. Mills will be able to attack in the future, is whether with a given operative temperature, that is with a given combination of air and wall temperature, it is better to have radiation or convection dominant. In brief, if you have hot walls and cold air, are there any physiological differences as compared with a situation in which you have cool walls and cool air, air and walls at about the same temperature but producing the same net thermal effect? That is the problem which we hope it may be possible to solve in the future.

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W. L. FLEISHER: The thing that impresses me in Dr. Mills' presentation of the paper is the fact that all of his experiments have been carried on with animals. Now, although I am not a doctor or a physicist, it is a well known fact that the emanation of heat from an animal, usually a fur-bearing animal, is principally sensible heat. It is interesting to note that with increased heat losses in human beings the latent heat loss increases much more rapidly than the sensible heat, and I bring it up simply as a question for Dr. Mills' consideration as to whether the experiments that he has carried on with radiant heat and cooling means would be similarly applicable with human beings. From the comfort angle which is involved in this, the necessity for the removal of large quantities of latent heat increases with the work performed and the conditions from which the people enter. In other words, the removal of the latent heat of perspiration is a very important function in establishing comfort. I feel that before we take too much for granted on the effect of radiant heat or radiant cold overcoming conditions which are established before the entrance into the particular enclosure, we must weigh the relationship between the sensible and latent heat losses of human beings and animals.

E. K. CAMPBELL: I had the pleasure of visiting Dr. Mills' laboratory on the way from Kansas City here, and I can testify that there is no effect on a human being as you enter those different rooms. The thought I want to bring out, however, is that we have a committee at the present time working on the problem of what we generally call intermittent heating, but in its essence it is exactly this same problem, the effect of the cold surface of a cold building when you have not heated anything but the air in the building and tried to make it comfortable. The problem involves the amount of heat, the rate of heat absorption necessary to overcome that absorption of radiant heat from the body within a reasonable length of time. The connection between the two has interested me very much.

ANSON HAYES:[2] I have been in close touch with Dr. Mills' work during the last year and think it very interesting. I thought it would be worthwhile perhaps to state what I consider a rational view of this field of development. Since there is a certain percentage of body heat loss that is effected by the radiant energy route, and since, depending upon the magnitude of that method of heat transfer, there will be the possibility of greater or less variation in air temperatures and humidities that can be tolerated with comfort by the individual, I think that this field of investigation should be approached from the standpoint that through the information that can be gained we will determine what that maximum variation in temperature of the air, with due consideration to the effect of relative humidity, may be and still have comfort by control of the radiant energy factor. It seems to me that is the broad problem that is involved.

J. DEB. SHEPARD: It has been interesting to hear the stress that Dr. Hick laid on the importance of radiant heat and the demonstration of its effects given in Dr. Mills' paper. I have personally found the conception of radiant heat effects a useful tool in handling practical conditions in air conditioning which result in complaints.

I will cite one case as an example, where it was found necessary to reduce the dry-bulb temperature in an enclosure below the usual design point to produce satisfactory comfort conditions due to the radiant heat from very heavy lens type illumination which was directed at the occupants of the space. There are many practical problems in both heating and cooling work, the solution of which is simplified by an understanding of radiant heat effects.

A. A. ADLER: The distinctions between conducted, convected and radiated heat are not as sharp as many authors seem to indicate. In the last analysis all heat energy is transferred by means of radiation with conduction and convection as special ensuing phenomena.

[2] *Director*, Research Laboratories, The American Rolling Mill Co., Middletown, Ohio.

Provost enunciated a theory of heat exchange which is substatially this: Two walls opposite each other at different temperatures radiate to each other irrespective of temperature. The colder wall radiates to the hotter and the hotter to the colder with the result that the hotter can radiate more than it receives because of the higher temperature. The net result is a transfer of energy to the colder wall. Equilibrium is reached when the mutual exchange is zero.

Conduction along a bar presupposes a temperature gradient. While the mutual exchange of heat energy along the bar is in accordance with Provost's theory, the hotter radiates more to the colder than it receives and energy is thus transferred.

In fluids, convection currents arise because heat energy enters molecules only by radiation and causes them to diminish their density. A current is set up, the denser pushing up the less dense. In all cases the only means for the transfer of heat is by electromagnetic radiation or simply radiation.

The broader aspect which includes selective radiation, absorption resonance phenomena at certain wave lengths, etc., need not concern us here. It is hoped that the authors and discussants may avoid any language which might indicate distinctions which do not actually exist.

DR. C. A. MILLS: Since you all had in your hands copies of this paper as it was published in November, I decided to use my allotted program time to show slides of the rooms and to present details omitted in the published article. If Mr. Fleisher will refer to the paper he will see there one whole section devoted to observations on human beings and will, I believe, find his question answered.

The only other discussions that need further comment are those by Professor Winslow, concerning *rediscovery*, and the wealth of knowledge he says is already available in this field. In presenting our preliminary paper on radiant conditioning, Dr. Ogle and I felt that we had two new contributions to make, contributions that had important bearings on interior conditioning and that had not previously been described in print. The first of these contributions was the demonstration that limitation of body heat loss almost exclusively to radiant channels was a safe, and apparently quite desirable, biologic step when tested through the whole life cycle of an animal. Comfort tests on human beings for a few hours are decidedly untrustworthy as a means of evaluating the biologic safety or desirability of any type of conditioning. Most of the present conditioning studies now being carried on, at New Haven and elsewhere, are of this type. I have repeatedly cautioned against interpretations based on such short-term observations.

To the best of my knowledge, we were the first to conduct longer-term conditioning studies on animals and demonstrate the biologic soundness of radiation cooling as a sole avenue of body heat loss. If Professor Winslow can cite references to previous work covering this point, I shall be delighted to give such workers proper credit.

The second of our contributions was the demonstration of *reflective* radiant conditioning, particularly *reflective radiation cooling*. It had been well demonstrated, at various times and places, that radiant conditioning could be accomplished through the use of hot or cold wall surfaces for heat radiation or absorption. Such systems possess numerous difficulties, both physical and economic, that have prevented practical application. Through the use of *heat-reflective* wall surfaces, we hoped to obviate most of these difficulties, as outlined in the paper. We had considered our findings as new and worthy of presentation. Again if Professor Winslow can cite references for his repeated use of the word *rediscovery* we shall be highly appreciative.

As regards the wealth of fundamental data in this field which Professor Winslow claims are so freely available, I do wish he would cite published studies on the transparency of pigment materials to heat wave lengths such as are radiated from the human body. We would like to have a color range of pigments for decorating

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the foil wall coverings, but the pigments must be transparent to these very long wave lengths. No physical tables so far found have carried any such data, nor have we found data showing whether foil reflects heat rays back at the same wave lengths possessed by those striking such surface. Basic studies are needed in this field. It is *not* a closed and finished chapter, as Professor Winslow's remarks would indicate.

Radiant heating studies in England have little bearing on the present discussion. In no case have they used *heat-reflective* wall surfaces as we are using. Their radiant heat makes a one-way trip across the room and then becomes kinetic heat that raises wall temperature. Our use of foil surface was for the purpose of minimizing this change from radiant to kinetic form except in the immediate vicinity of the individual who is being conditioned. Any other system entails great waste of heating or cooling load on air mass and building materials.

I cannot agree with Professor Winslow that insulation will still be needed in *reflective radiant conditioning*. Foil surfaces maintain themselves at contact air temperatures, playing an entirely passive reflective role in the system. Because of the low emissivity of foil surface to heat within the foil itself, its surface acts as a bar to radiation *into* the room of heat coming through the wall. It will, it is true, readily give up such heat to the air in direct contact with its surface, but air temperatures within and without the room should be at approximately the same level with radiant conditioning in summer heat.

Much remains to be done to bring our ideas in this field into form for practical use. It is hoped that a field test of the ideas can be arranged in the near future.

No. 1081

PHYSIOLOGICAL REACTIONS AND SENSATIONS OF PLEASANTNESS UNDER VARYING ATMOSPHERIC CONDITIONS

By C.-E. A. WINSLOW * (*MEMBER*), L. P. HERRINGTON,** AND A. P. GAGGE *** (*NON-MEMBERS*), NEW HAVEN, CONN.

THE aim of the art of air conditioning (aside from certain special industrial applications) is to produce conditions which are healthful and pleasant for human beings; and it has been made clear, by the researches of the past quarter-century, that the major effects of the atmospheric environment upon health and comfort are directly related to thermal interchange between the human body and that environment.

OBJECTIVES OF STUDY

A comprehension of the nature of these interchanges is essential to sound progress in heating and ventilating practice. Yet the contributions of physiology on this point have been meager. Pettenkofer and Rubner, in the Nineteenth Century, made a good beginning but the leads they furnished have not been followed up, and in the literature of physiology there are practically no fundamental analyses of this problem based on serious experimental attempts to obtain simultaneous measurements of the various factors involved under a wide range of thermal conditions. The most significant contributions have been made by engineers who have boldly and successfully entered the physiological field and among them, the A.S.H.V.E. studies [1] stand preeminent. Largely on the basis of the experiments of these investigators at Pittsburgh, Missenard,[2] an engineer in France, has developed a valuable theoretical analysis of the problem.

Contribution No. 21, from the John B. Pierce Laboratory of Hygiene.
* Director, John B. Pierce Laboratory of Hygiene.
** Associate Director, John B. Pierce Laboratory of Hygiene.
*** Physicist, John B. Pierce Laboratory of Hygiene.

[1] Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. Ed. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, 1929, Vol. 35, p. 245).

[2] Etude Physiologique et Technique de la Ventilation, by F.-A. Missenard, L. Eyrolles, Paris, 1933.

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the Joint Session with the *A.S.R.E.*, New York, N. Y., January, 1938, by C.-E. A. Winslow.

The A.S.H.V.E. studies did not permit of the separate evaluation of the influence of radiation. It has been the aim of the work at Pierce Laboratory to include this important factor and to attempt an exact analysis of the influence

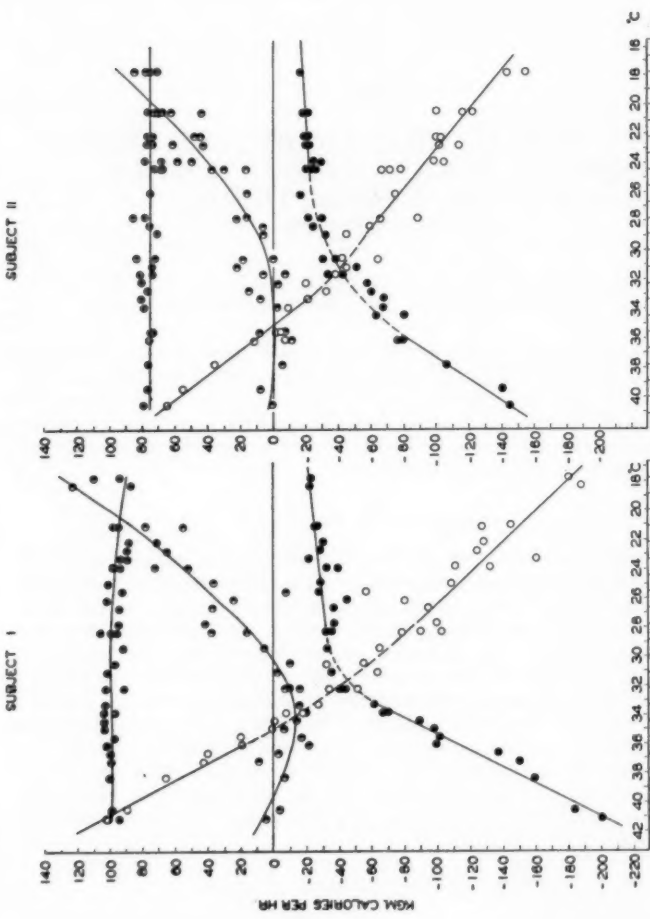

FIG. 1. ELEMENTS OF HEAT PARTITION FOR A STOUT (I) AND A SLENDER (II) SUBJECT AT VARYING OPERATIVE TEMPERATURES (T_o)

upon physiological reactions and human comfort of widely varying conditions of air temperature, wall temperature, air movement, and humidity, conducted with the most precise physical instruments available and under conditions which make possible the simultaneous quantitative determination of all the thermal processes involved.

EXPERIMENTAL METHODS EMPLOYED

For this purpose there has been devised a new method of study which was named *partitional calorimetry* (Winslow, Herrington and Gagge).[3] The subject under observation is placed in a copper booth, from the walls of which radiant heat may be reflected in any desired degree without producing convection currents; the temperature, movement, and humidity of the air being controlled by the usual procedures. Metabolism is determined by the ordinary methods. Evaporation from the body is measured directly by weight loss recorded on a delicate scale upon which the subject is seated. Gain or loss of heat by convection is computed from temperature differences between the body surface and the surrounding air; and gain or loss by radiation from temperature differences between the body surface and the surrounding walls. Mean body surface temperature is derived from a weighted average of thermopile observations at 15 points on the body surface. Storage (cooling or heating of the body tissues) is determined from changes in rectal temperature within a range where such determinations are representative; under cold conditions, where the rectal temperature fails to keep pace with cooling of the tissues, storage is obtained indirectly from the difference between the metabolism and the other three factors as directly measured.

Evidence has been presented to show that results obtained in this way and based on the mean of five duplicate experiments have an error of less than 5 per cent (Winslow, Herrington and Gagge[4]).

All of the data discussed in the present paper refer to unclothed subjects in a semi-reclining position. The absolute values cited do not, therefore, apply to ordinary conditions of practice. The fundamental principles, however, must underlie any valid analysis of the relations between the human body and its thermal environment. Studies on clothed subjects are now being carried forward in the laboratory.

BASIC EQUATIONS AND CONSTANTS INVOLVED

The fundamental thermodynamic processes concerned may be described by the equation:

$$M + S = E + R + C;$$

where

M = rate of metabolism,
S = rate of storage,
E = rate of evaporative heat loss,
R = rate of radiative heat loss or gain,
C = rate of convective heat loss or gain.

All units are expressed in kilogram-calories per hour, and storage is considered positive when the body cools, negative when the body becomes warmer.

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[3] A New Method of Partitional Calorimetry, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal Physiology*, 1936, Vol. 116, p. 641).

[4] The Determination of Radiation and Convection Changes by Partitional Calorimetry, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal Physiology*, 1936, Vol. 116, p. 669).

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R and C are positive when T_S is greater than T_W and T_A, respectively; negative when T_S is less than T_W and T_A.

Metabolism and evaporative heat loss are determined by direct experimental methods. Storage, as pointed out heretofore, is either calculated from the rate of change in body temperature or computed by difference.

Radiation interchange is governed by the Stefan-Boltzmann Law, and expressed by the equation,

$$R = A_R k (T_W^4 - T_S^4);$$

or

$$= A_R (4k\ T_S^3)\ (T_W - T_S), \text{ as a first approximation,}$$

where

R = radiation exchange,
A_R = effective radiation area of the body in square meters,
k = the universal radiation constant,
T_W = mean black body radiant temperature of environing surface in absolute degrees,
T_S = mean skin temperature in absolute degrees.

The unknown factor here is A_R, the effective radiation surface of the body. This will, of course, be less than the total surface of the body and will vary with body build and position during the experiment. It has been necessary, therefore, to determine this factor for each individual, which can readily be done by holding the difference between skin and air temperature constant (which holds convection loss also constant) and varying wall temperatures (Gagge[5]).

Convection loss will depend upon mean skin temperature, air temperature, and air movement. The constants involved in convection loss vary with the form of the object cooled and even the type of formula which applies to the phenomenon of air movement is not agreed upon.

The formula most commonly used is as follows:

$$C = A\ (\triangle\ T)^a + B\ (\triangle\ T) V^b,$$

where $\triangle\ T$ represents the difference between skin and air temperature, and *where* A, B, a and b, are constants. For such limited air velocities as occur under ordinary conditions of ventilation, the exponential constants may be considered as equaling unity. The formula then becomes:

$$C = (A + B\ V)\ \triangle\ T.$$

Other students of the problem have employed a quite different relation:

$$C = A \sqrt{V}\ \triangle\ T.$$

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[5] The Linearity Criterion as Applied to Partitional Calorimetry, by A. P. Gagge (*American Journal Physiology*, 1936, Vol. 116, p. 656).

Here there is but one arbitrary constant and convection loss is assumed to vary with the square root of air velocity.

The authors' studies [6] indicate that either of these two formulae will describe observed results with equal accuracy when air velocity is not very high. If the entire curve is parabolic, the authors' experiments are obviously in a section of the curve which is essentially linear. Therefore, the square root type of formula has been used as equally accurate and simpler of application, and A has been determined for each subject at varying air temperatures.

The fundamental partitional equation, then, assumes the following form:

$$M + S = E + A_R K (\triangle T_W) + A_C (\triangle T_A) \sqrt{V},$$

where

K = $(4kT_S^3)$ from the first approximation of Stefan's Law,[7]
A_R = the effective radiation area for a given subject,
$\triangle T_W$ = difference between skin and wall temperatures,
A_C = convection constant for a given subject,
$\triangle T_A$ = difference between air and skin temperatures,
V = air velocity.

GENERAL RELATIONS OBSERVED

To simplify analysis of the results obtained, first consideration shall be given the effects of radiation and convection as combined, and the thermal interchanges shall be partitioned under the four headings of metabolism, evaporation, storage, and radiation plus convection. The distinctive influence of radiant heat will be examined in a later paragraph. For such an analysis, the figures must obviously be compared for energy interchange with a temperature scale which represents the combined influence of both radiation and convection; and for this purpose there has been introduced the concept of *operative temperature*. *Operative temperature* is the temperature which represents the net physical effect of a given combination of air and wall temperatures (Winslow, Herrington and Gagge [8]; Gagge, Herrington and Winslow [9]).

The general results obtained with two subjects are presented in Fig. 1. Subject I was an extremely stout individual and Subject II a very slender one, representing the extremes of ordinary body build. The velocity of turbulent air movement in the experiments plotted here was 15-20 linear feet per minute, and relative humidity between 40 and 50 per cent. It will be noted that operative temperatures vary from 64 to 108 F. These figures represent the resultant of much more widely ranging air and wall temperatures. Actually, air temperatures varied from 43 to 97 F and wall temperatures from 66 to 135 F. In most experiments the walls were warmer than the skin and the effect of

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[6] Thermal Interchanges Between the Human Body and Its Atmospheric Environment, by A. P. Gagge, L. P. Herrington and C.-E. A. Winslow (*American Journal of Hygiene*, 1937, Vol. 26, p. 84).

[7] For purposes of analysis, K may be evaluated as a constant by assuming a mean value of T_S over the range of experiment.

[8] Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal Physiology*, 1937, Vol. 120, p. 1).

[9] Loc. Cit. See Note 6.

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radiation ranged from a loss of 84 to a gain of 189 kilogram-calories per hour. The air was generally cooler than the skin and the effect of convection ranged from a loss of 183 to a gain of 4 kilogram-calories per hour. The following phenomena are obviously apparent in both graphs.

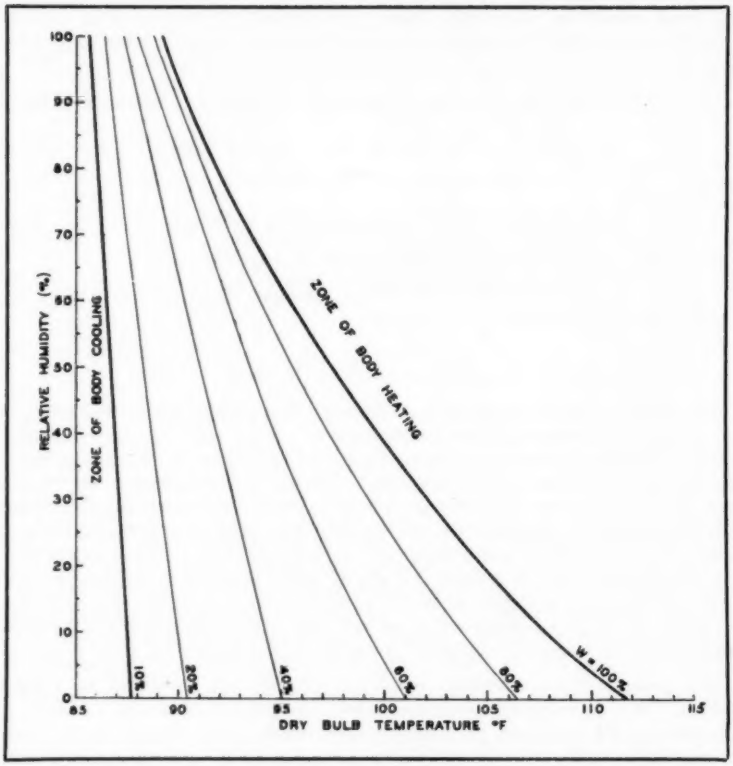

FIG. 2. CONTOUR CHART INDICATING VARIATIONS IN WETTED AREA (w) WITH VARYING AIR TEMPERATURES AND RELATIVE HUMIDITY WITHIN THE ZONE OF EVAPORATIVE REGULATION

1. Metabolism (for a given subject) remained approximately constant within the range of operative temperatures employed. This would not, of course, be the case under more extreme conditions.

2. At a critical temperature of 88 to 90 F the heat produced by metabolism (roughly 100 kilogram-calories per hour for Subject I) is balanced by the heat loss due to evaporation and to radiation plus convection—these two major components accounting for about 50 kilogram-calories each. Storage is zero; that is, the body tissues show no change in temperature.

3. As one proceeds to higher operative temperatures, the heat received from the

environment due to radiation plus convection increases progressively but is exactly balanced by a similarly progressive increase in evaporative heat loss, so that no negative storage (warming of the body) takes place. At an operative temperature of 106 F the body is gaining 100 kilogram-calories from the combined influence of walls and air, and this heat gain plus the metabolic heat produced is balanced by a heat loss of 200 kilogram-calories due to evaporation.

4. Below the critical temperature of 86 F the phenomena are wholly different. Evaporative heat loss here changes but slightly, falling from 40 to 20 kilogram-calories per hour as the operative temperature decreases, as a result chiefly of the

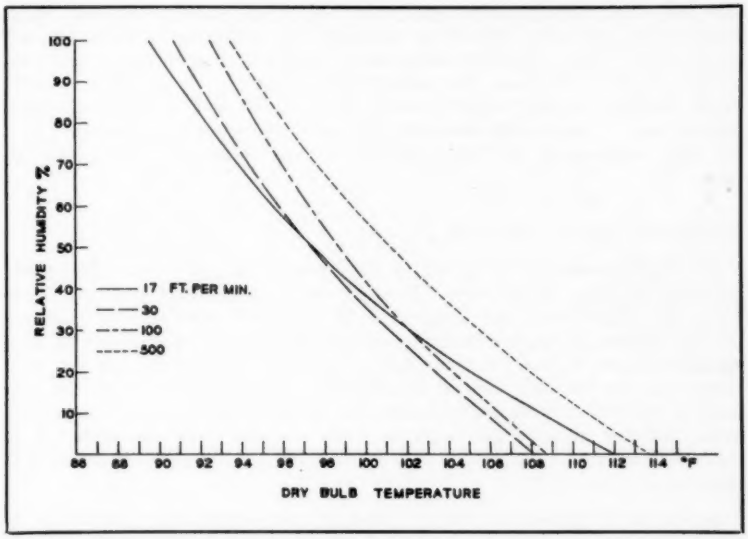

FIG. 3. CONTOUR CHART INDICATING UPPER LIMITS (WETTED AREA = 100 PER CENT) OF THE ZONE OF EVAPORATIVE REGULATION FOR VARIOUS AIR VELOCITIES

purely physical factor of decreasing vapor pressure difference between skin and air. In this zone, heat loss due to the combined influence of radiation plus convection increases progressively (although the slope of the line is less rapid than above 86 F for reasons to be discussed in a later paragraph). Since this progressively increasing heat loss is not balanced, there is a parallel increase in storage (cooling of the body tissues).

It is clear that no simple equation can describe the influence of atmospheric conditions above and below the critical point in these graphs since the relative influence of air temperature, air movement and relative humidity will be quite different on the two sides of the inflection point. The area above 88 F has been called the Zone of Evaporative Regulation, and the area below 86 F the Zone of Body Cooling, and the phenomena apparent in each of these zones must be considered separately.

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REACTIONS IN ZONE OF EVAPORATIVE REGULATION

In this area, it has been observed that the effects of an increasingly warm environment are exactly balanced by a parallel increase in evaporative cooling. At low operative temperatures (below 81 F) a constant evaporative loss of about 30 kilogram-calories per hour is observed, which represents minimum evaporation from the skin in the absence of sweat secretion. As operative temperature rises in the Zone of Evaporative Regulation, skin temperature also rises and when the skin temperature reaches 93 F active sweat secretion begins. Above this point skin temperature remains approximately constant but with progressively rising operative temperature sweat secretion increases in exactly parallel fashion (Winslow, Herrington and Gagge [10]). To express the effect of the increase in sweat secretion there has been derived a new physiological constant *wetted area* which describes the extent of moisture surface present on the total surface of the body (Gagge [11]; and Winslow, Herrington and Gagge [12]).

Influence of Relative Humidity

The experiments plotted in Fig. 1 were conducted with a constant relative humidity of 40 to 50 per cent. In other studies this factor has been varied (Winslow, Herrington and Gagge [13]). It has been found that the body can balance changes in relative humidity within this zone just as precisely as it can balance changes in temperatures—up to the point where the sweat no longer evaporates but runs off without exerting its cooling power. Until the latter point is reached, evaporative heat loss at a given air temperature is exactly the same, whether humidity be high or low. As the vapor pressure of the atmosphere increases, the wetted area increases *pari passu* so as to maintain the required heat loss.

In Fig. 2 are presented curves indicating how wetted area changes in this zone with varying air temperature and relative humidity (the walls being at the same temperature as the air in all cases). The 10 per cent line (wetted area equal to 10 per cent of the potentially wettable area) corresponds to the minimum evaporation occurring without active sweat secretion. The 100 per cent line corresponds to the condition beyond which sweat no longer evaporates but collects and runs off the body without exerting its cooling power. Above the points on this line the process of evaporative cooling fails and the body temperature must rise.

Influence of Air Movement

The discussion in preceding paragraphs and the data presented in Fig. 2 refer to conditions in which there was a turbulent air movement of 17 linear feet per minute. With variation in air movement, the rate of evaporation and

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[10] Loc. Cit. See Note 8.
[11] A New Physiological Variable Associated with Sensible and Insensible Perspiration, by A. P. Gagge (*American Journal Physiology*, 1937, Vol. 120, p. 277).
[12] Physiological Reactions of the Human Body to Various Atmospheric Humidities, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal Physiology*, 1937, Vol. 120, p. 288).
[13] Loc. Cit. See Note 12.

the wetted area will change in accordance with formulae which have been presented elsewhere (Gagge, Herrington and Winslow [14]). The effect of this factor can best be understood by analyzing its influence upon the 100 per cent line of Fig. 2, representing the upper limits of possible evaporative regulation. This influence is illustrated in Fig. 3.

It will be noted that (for the nude subject in a semi-reclining posture) increase in air movement consistently increases evaporative cooling, and there-

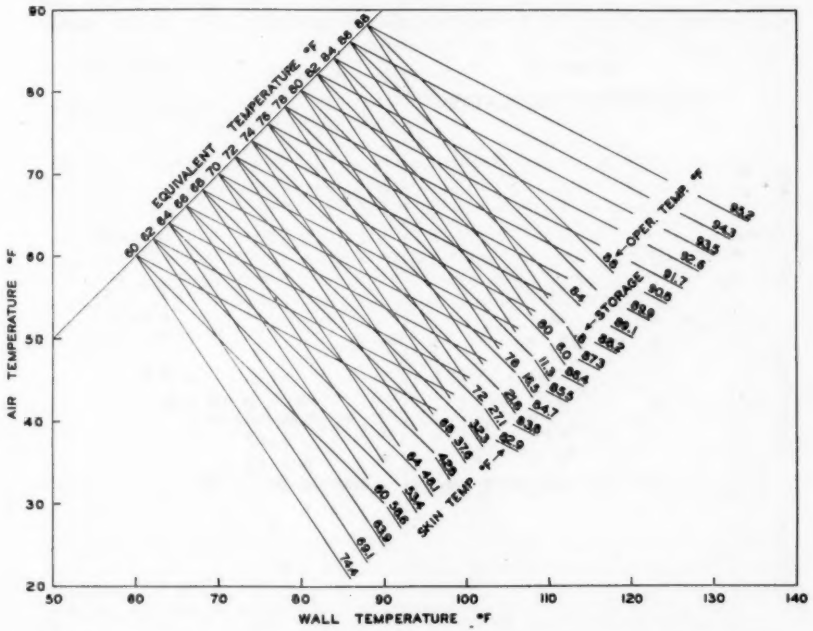

FIG. 4. CONTOUR CHART FOR PREDICTION OF SKIN TEMPERATURE AND STORAGE FOR VARIOUS COMBINATIONS OF AIR TEMPERATURE AND WALL TEMPERATURE (RELATIVE HUMIDITY 40-50 PER CENT; AIR MOVEMENT, 17 FPM)

fore heat tolerance, when relative humidity is high and air temperature low. When relative humidity is low and air temperature is high, however, an increase in air velocity from 17 to 30 or 100 linear feet per minute actually decreases heat tolerance. Under these conditions, air temperature is above skin temperature and the increased demand for evaporative cooling exceeds the actual increase of evaporation due to the higher air movement. When an air velocity of 500 linear feet per minute is reached, the limits of the zone are broadened throughout its range. For very hot and dry environments, still air is more

[14] Loc. Cit. See Note 6.

desirable than a slightly greater air movement but a very high air movement is even better.

REACTIONS IN ZONE OF BODY COOLING

In this area (below 86 F for the unclothed body), the chief changes which occur, as operative temperature decreases, are (*a*) increased heat loss due to radiation plus convection, and (*b*) increased positive storage or cooling of the body tissues. It will be noted by reference to Fig. 1 that the slope of the line representing heat loss due to radiation plus convection changes as one passes

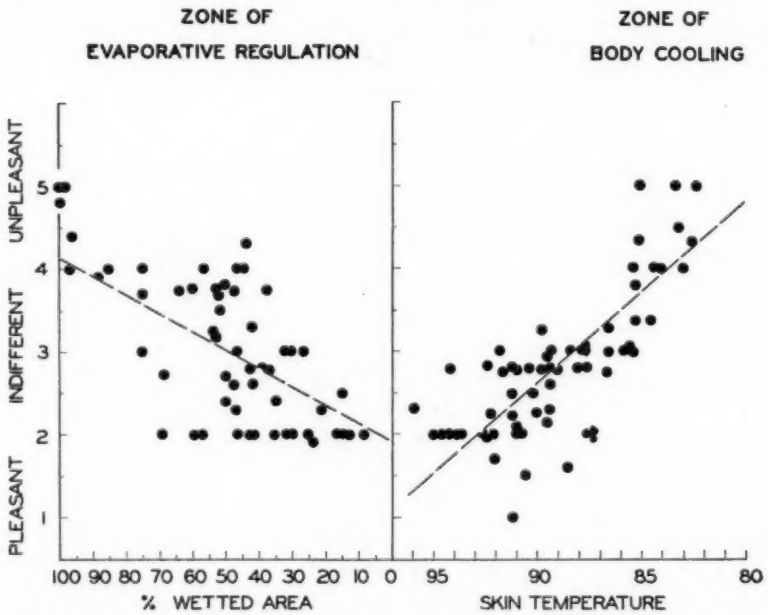

FIG. 5. RELATION OF SENSATIONS OF PLEASANTNESS TO WETTED AREA (IN THE ZONE OF EVAPORATIVE REGULATION) AND TO SKIN TEMPERATURE (IN THE ZONE OF BODY COOLING)

from the Zone of Evaporative Regulation to the Zone of Body Cooling. The less abrupt slope in the latter zone is due to a progressive fall in skin temperature which is the only mechanism the body calls into play in this region to adapt to a cool environment. Under colder conditions, or after longer periods of time, a second mechanism—increased metabolism—would become operative but this does not appear in the experiments here reviewed.

For a fall in operative temperature from 88 to 68 F the mean skin temperature decreases from 94 to 84 F. The temperature of the lower extremities falls most rapidly while that of the head or trunk may decrease less. In general,

however, all areas of the skin tend to maintain a differential of at least 8 F above the environment. There has been derived a physical factor, conductance, which measures the result of this adaptive variation, expressed in terms of the rate at which heat passes from the deeper tissues of the body to its surface. It depends on a constant factor of tissue conductivity for the individual and on the state of dilation of the superficial blood vessels (Winslow, Herrington and Gagge [15]).

This type of regulation is, however, as pointed out previously, incomplete; and positive storage (cooling of the body) increases progressively as air temperature falls. Fig. 4 shows the effect upon skin temperature of various combinations of air and wall temperature for the unclothed body (Gagge, Herrington and Winslow [16]).

Influence of Air Movement

It is, of course, obvious that air movement will always increase heat loss due to convection, while evaporation (which is quantitatively unimportant in this zone), will not be greatly influenced. Radiation will not be directly affected but radiative heat loss to cold walls will be decreased by the fall of skin temperature which results from greater convective loss, and radiative heat gain from hot walls correspondingly increased. The relation of storage to air and wall temperatures for various air velocities has been derived as follows:

For a velocity of

$$\begin{aligned}
&17 \text{ fpm; } S = 232.2 - 1.03\ T_A - 1.60\ T_W,\\
&28 \text{ fpm; } S = 246.0 - 1.22\ T_A - 1.55\ T_W,\\
&50 \text{ fpm; } S = 276.0 - 1.60\ T_A - 1.50\ T_W,\\
&100 \text{ fpm; } S = 290.0 - 1.74\ T_A - 1.45\ T_W,
\end{aligned}$$

where S equals storage in kilogram-calories per square meter of body surface per hour.

It should be made clear that storage, as defined by the authors, is a measure of the deviation of the body from thermal equilibrium after a specified time of exposure (in these experiments, one-and-a-half hours). It does not measure the total heat loss during this period but the rate of heat loss at the end of it.

RADIATION AS DISTINGUISHED FROM CONVECTION

So far, heat losses and heat gains due to radiation and convection have been considered, as if their influence upon the body corresponded entirely to their potency as measured by ordinary physical instruments. This is the case in the Zone of Evaporative Regulation (where no significant changes in skin temperature are manifest). In the Zone of Body Cooling, however, distinctly different physiological reactions to the two forms of heat interchange may be observed (Herrington, Winslow and Gagge [17]).

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[15] Loc. Cit. See Note 8.
[16] Loc. Cit. See Note 6.
[17] The Relative Influence of Radiation and Convection Upon Vasomotor Temperature Regulation, by L. P. Herrington, C.-E. A. Winslow and A. P. Gagge (*American Journal Physiology*, 1937, Vol. 120, p. 133).

If two conditions are compared in which the same operative temperature is maintained—by warm walls and warm air and by hot walls and cold air respectively—markedly different responses are found in skin temperature (and hence in conductivity of the superficial body tissues). With cold air and hot walls

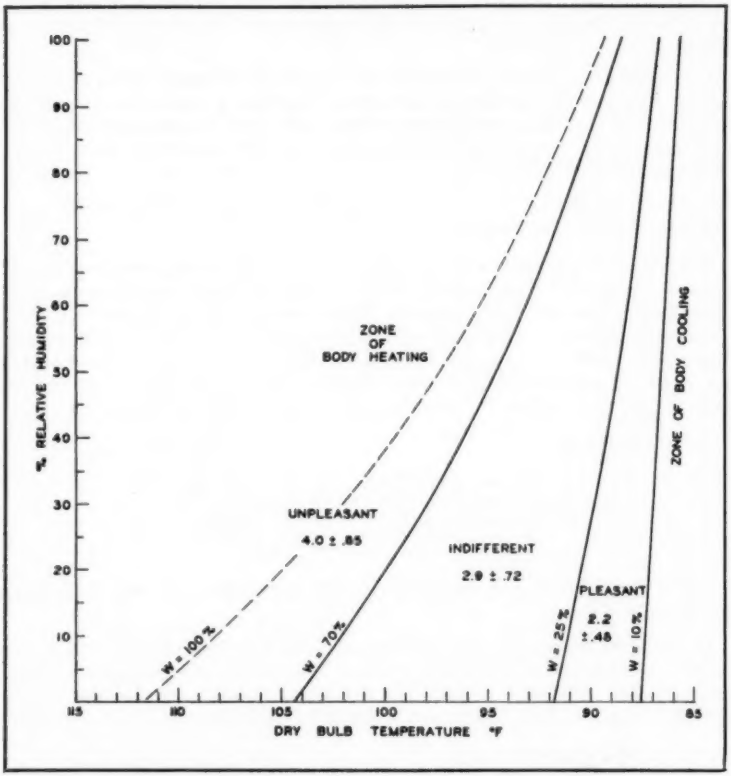

FIG. 6. CONTOUR CHART INDICATING LIMITING WETTED AREAS ASSOCIATED WITH CERTAIN SENSATIONS OF PLEASANTNESS (IN THE ZONE OF EVAPORATIVE REGULATION) IN RELATION TO AIR TEMPERATURE AND RELATIVE HUMIDITY. OBSERVED MEAN COMFORT VOTES ARE INDICATED IN EACH REGION

the skin temperature falls more rapidly than with air and walls, of more nearly equal temperature, exerting the same total thermal effect from a physical standpoint.

If metabolism and evaporation remain normal, there is found the following prediction equation, indicating the greater influence of air than wall upon skin temperature:

$$T_B = 029.\ T_A + 0.15\ T_W + 56.5.$$

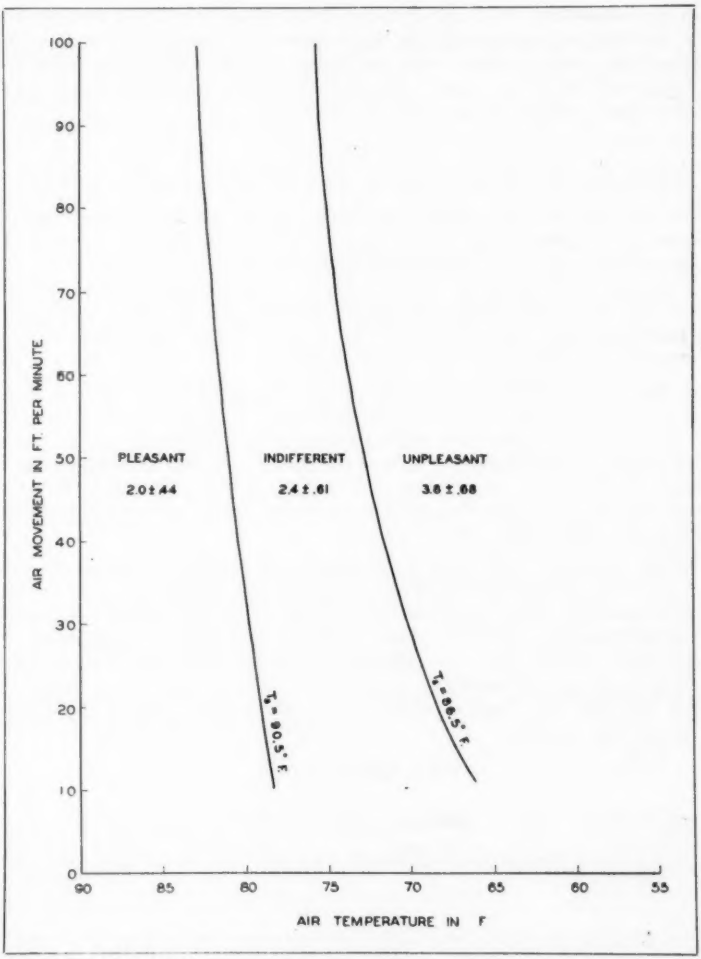

FIG. 7. CONTOUR CHART INDICATING LIMITING SKIN TEMPERATURES ASSOCIATED WITH CERTAIN SENSATIONS OF PLEASANTNESS (IN THE ZONE OF BODY COOLING) IN RELATION TO AIR TEMPERATURE AND AIR MOVEMENT. OBSERVED MEAN COMFORT VOTES ARE INDICATED IN EACH REGION

Whether this differential effect is due to any specific differences in the effect of radiation and convection upon the skin or merely to the fact that heat loss from the respiratory tract and that due to increased convection loss by slight body movements are governed even by air temperature is not known. It is clear, however, that with warmer walls and cooler air the body adapts itself more readily to cold conditions than it does to a thermally-equivalent environment in which air and walls are at the same temperature.

SENSATIONS OF PLEASANTNESS IN ZONE OF EVAPORATIVE REGULATION

In connection with the studies here discussed, the subjects were asked at the close of each experiment to record their subjective impressions with regard to the atmospheric conditions maintained, on a scale of five points, as follows: (1) very pleasant; (2) pleasant; (3) indifferent; (4) unpleasant; and (5) very unpleasant. These expressions of opinion proved to be correlated with physical conditions and physiological reactions to an even higher degree than had been anticipated (Winslow, Herrington and Gagge [18]).

At the point of thermal equilibrium (operative temperature between 88 and 90 F) there were eight groups of experiments. In five of these groups, the vote was 2 (pleasant) or less; in two groups it was between 2 and 2.5; in one group it was 3 (indifferent). This is the range in which skin temperature is normal (88-93 F), storage minimal and evaporation low. On either side of this region votes became progressively higher (more unpleasant) when proceeding to hotter or colder conditions.

On the hot side, in the Zone of Evaporative Regulation, significant correlations were found between sensations of pleasantness on the one side and both skin temperature and wetted area on the other. When wetted area is held statistically constant, however, the correlation with skin temperature becomes insignificant; while, when skin temperature is held constant, the correlation between sensation and wetted area remains $+0.51 \pm 0.08$. It is found that, within this zone, the relation of votes as to pleasantness may be expressed by the prediction equation:

$$C = 0.022\, w + 1.95,$$

where

C = sensations of comfort or pleasantness, with high value indicating unpleasantness.
w = wetted area.

The relations are graphically indicated in the left half of Fig. 5.

In this Zone, at a given rate of air movement, wetted area (and hence the sensation of pleasantness) is determined by relative humidity and operative temperature (or air temperature if air and wall temperature are the same). The range of sensation votes for the latter condition, with respect to air tem-

[18] Relations Between Atmospheric Conditions, Physiological Reactions and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Hygiene*, 1937c, Vol. 26, p. 103).

perature and relative humidity, is indicated in Fig. 6. This graph is based on an air movement of 17 linear feet per minute. Higher air movement will modify the effects as noted in an earlier paragraph.

It is interesting to note that throughout the Zone included between the lines of Fig. 6, thermal regulation is perfect. Sweat secretion is successfully balancing the warmth of the environment; but this process—in spite of the fact that sweat is evaporating as fast as it is formed—is accompanied by marked sensations of unpleasantness. For an air movement of 17 linear feet per minute, the degree of sweat secretion associated with sensations of distinct unpleasantness will occur beyond the following limits of temperature and relative humidity:

TEMPERATURE, F	RELATIVE HUMIDITY, PER CENT
89	96
92	70
95	48
98	30
101	19
104	0

SENSATIONS OF PLEASANTNESS IN ZONE OF BODY COOLING

In this region widely different phenomena are observed. Here, there is a small correlation between sensations of pleasantness and storage, which is not statistically significant, and a highly significant correlation between sensations of pleasantness and skin temperature (-0.64 ± 0.09). The general relationship is indicated on the right half of Fig. 5.

It is found that, in this area, the pleasantness of the thermal environment may be predicted from the relation:

$$C = (-0.22)\, T_S + 22.4.$$

The chief variables affecting skin temperature (and hence sensation) in this Zone are operative temperature (or air temperature, if air and wall temperatures are the same) and air movement. The limits for various degrees of pleasantness or unpleasantness are indicated in Fig. 7.

Where air and wall temperatures differ, it has been pointed out that the influences upon skin temperature of radiant heat and convected heat are different. With cold air and warm walls, skin temperature will be lower and discomfort greater than with the same operative temperature produced by walls and air of identical temperature.

The following conditions represent limits beyond which distinct sensations of unpleasantness will occur with different combinations of air and wall temperature and with air velocities of 17 and 100 linear feet per minute:

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AIR MOVEMENT 17 FPM		AIR MOVEMENT 100 FPM	
Air Temp. F	Wall Temp. F	Air Temp. F	Wall Temp. F
68	68	76	76
65	73	73	91
62	80	70	105
59	86	67	118

LIMITATIONS OF THE PRESENT STUDY

It cannot be too clearly emphasized that the results of the studies here reported refer only to the unclothed subject, at rest in a semi-reclining position. It is believed that the principles developed are of general application but the absolute values derived apply only to the conditions of the experiments.

These data cannot, therefore, be directly compared with those of the A.S.H.V.E. Comfort Chart (THE GUIDE, 1937). Other subjects were always unclothed except for an athletic supporter, while the Pittsburgh subjects (McConnell, Houghten[19]; Houghten and Yagloglou[20]) were fully clothed under cold conditions, and stripped to the waist only, under hot conditions. The present subjects were at rest in a semi-reclining position while the Pittsburgh subjects were allowed a certain freedom of movement. Metabolisms were, therefore, substantially higher than those in the tests herein discussed (Houghten, Teague, Miller and Yant[21]) so that limiting temperatures for unpleasant conditions on the cold side were also inevitably higher than those of the present authors.

It is a little puzzling that the slope of the lines in Fig. 6 is substantially different from what would be expected from the lines of the Comfort Chart, the influence of relative humidity seeming to be less marked in present studies. Any detailed analysis of such differences can profitably be postponed until the work now being done with clothed subjects is complete.

DISCUSSION

PROF. C. P. YAGLOU (WRITTEN): This work of Professor Winslow and associates cannot be praised too highly. It is a most thorough physiologic evaluation of thermal factors involved in human comfort and well being. Although the results presented apply to unclothed persons only, the fundamental principles adduced are of great value in clarifying the mechanism of response to radiant heat and atmospheric humidity. Of much greater interest should be the results with clothed subjects. It is hoped that at the completion of this work we will have sufficient basic data for assessing comfortable and pleasant environments in terms of component factors,

[19] Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, 1923, Vol. 29, p. 129).
[20] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yagloglou (A.S.H.V.E. TRANSACTIONS, 1923, Vol. 29, p. 361).
[21] Loc. Cit. See Note 1.

or perhaps in terms of some composite index, like the effective temperature index, suitably modified to include the radiation component.

T. W. CARRAWAY:[22] In attempting to follow Professor Winslow, I noticed that several actions were subject to a like reaction, which, of course, follows an old rule, and it leads me to ask the question: Dr. Winslow, have you conducted tests as to the extent of determining whether the human being becomes sensitive or immune to the temperature difference that would cause a stifling feeling or even nausea? May I make myself perhaps a little clearer by saying that in the event the body was exposed to a condition that would cause this distress, by repetition would it become sensitive or would it become immune?

PROF. C.-E. A. WINSLOW: We have not done any experiments on exactly that line, but I should feel pretty sure that such would be the case because there has been much work done, particularly by the English observers, on the relation of comfort and efficiency to acclimatization. They found that the temperature for maximum comfort and maximum efficiency in different seasons of the year might vary as much as five degrees with the extent to which the people became acclimated to that situation. I think we are all familiar with the let-down that you feel on the first warm days in spring while similar days in summer would feel very comfortable. Those feelings are a reflection of a better physiological adjustment that is attained after exposure to hot or cold conditions.

W. H. CARRIER: I might add another confirmation to what you have just said, Professor Winslow. In India a cotton mill was reduced in temperature to 80 deg and 80 per cent humidity. This was much lower than they had been accustomed to. They had 110 or 105 outside. The operatives all struck; it was too cold. We here would think that it was very comfortable or perhaps too hot in a fairly unclothed condition such as they were working in.

J. J. AEBERLY: I believe that a paper as well prepared as this one should be more freely discussed. The presentation of the facts by Professor Winslow was well done. The physiological aspect of the problem has been handled in a way as to be easily understood by engineers.

The situation in general, regarding the program of study by our Society treating with the physiological response of man to his environment, may be likened to a routine which a mathematician must follow to find the center of mass enclosed in a warped surface. He must determine the perpendiculars to the warped surface and locate the points of intersections, ultimately solving his problem.

We, too, by means of study such as this, must find the center of mass or, in other words, the basic fact or facts underlying man's response to his environment. After these facts are established, we shall have a greater assurance that the designs properly incorporating these findings will stand test.

PROFESSOR WINSLOW: I am very glad Mr. Aeberly emphasized this last point, and I want to explain it in just a word. These results we have given today refer only to the unclothed subject in a semi-reclining position. The whole thermal problem changes when the subject is clothed. It changes when the subject is moving about with greater metabolic activity. It changes with age and sex and with health.

PROFESSOR YAGLOU: I have the highest praise for the work which Professor Winslow and his associates are doing at Yale. More important is the work that is yet to come with the subjects fully clothed, and I am looking forward to it with great interest.

W. L. FLEISHER: Speaking for the Research Committee of the Society, on which we are very fortunate in having Professor Winslow, and also to make a statement

[22] Pres., Carraway Engrg. Co., Dallas, Tex.

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that the three papers delivered today on the physiological reactions of human beings are all in a way correlated researches or uncorrelated researches rather of the Research Laboratory. It is our hope in the coming year, through our members of the Research Committee and our very well-correlated activities, to bring together the various divergent ideas which may exist in the various views that have been expressed by the members that have spoken both this morning and this afternoon.

No. 1082

PERFORMANCE TESTS OF ASBESTOS INSULATING AIR DUCT

By R. H. Heilman * (*MEMBER*), Pittsburgh, Pa., and R. A. MacArthur ** (*NON-MEMBER*), Lockland, O.

THE all-asbestos duct is a new development in the air conditioning field although asbestos insulation has been used extensively on metal ducts. Data are presented briefly in this paper to indicate the suitability of asbestos for the fabrication of air conditioning ducts of satisfactory engineering characteristics. Friction losses, acoustic characteristics, and thermal data are presented and discussed.

Description of Asbestos Ducts

Asbestos ducts are of double-shell construction and at present are available in lengths of 3 ft. The stiff inner shell is approximately 3/16 in. in thickness and is made from hardened laminated asbestos board. The outer shell, or jacket, makes a close sliding fit with the inner shell, or core, and consists of layers of corrugated or indented asbestos paper, having thermal insulating value, lined and covered with protecting hardened laminated asbestos paper. The standard over-all thicknesses are ½ in. and 1 in.

Unretouched photographs of two sections of standard thicknesses are shown in Fig. 1. The inner shells have been partially withdrawn from the outer shell jackets.

For assembly of units into ducts, an inner shell is withdrawn as shown in Fig. 1 for a distance of, conveniently, 8 in. The extended core is telescoped into the jacket of another unit. This is continued until the required length is obtained. Adhesive tape or metal bands may be applied to the joints where the duct is exposed to view as the joint cracks would give an unsatisfactory appearance. A carpenter's saw is used for cutting cores or shells for fitting. The stiffness of these ducts is greater than that of sheet metal of usual gages, and the joints retain their integrity under considerable abuse. Ells and turns are assembled with asbestos directing vanes held in position with metal clips of special design. Figs. 2 and 3 indicate the method of fabrication of 90 deg ells. Tees are assembled and branches are taken from trunk ducts by combinations of straight lengths and turns, as indicated in Fig. 4. Sizes of ducts are estab-

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lished so that transformations, as indicated in Fig. 4, are possible over a wide range of size combinations.

APPARATUS AND TESTS

Most of the data are presented from the practical viewpoint. For the most part, friction losses were directly measured for various air velocities and sizes of ducts by exploration of duct cross-sections with a portable air meter having a sensitivity of 0.005 in. A few static pressure drops were obtained for various lengths of ducts by means of water-filled manometers reading plenum chamber static pressures, velocities being obtained in the ducts by means of a Pitot head and gage.

An attempt was made to determine the practical value of asbestos ducts in the reduction of sound in installations made therefrom. The sound measure-

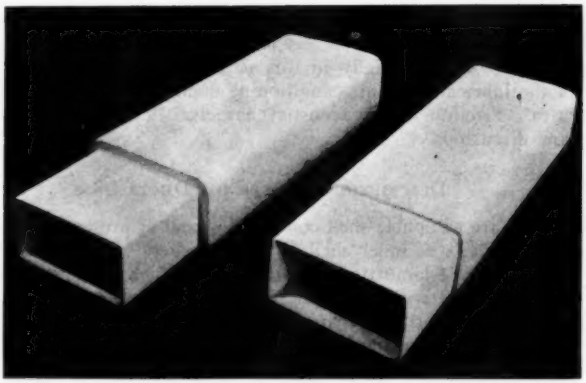

FIG. 1. ASBESTOS DUCT WITH INNER SHELL PARTIALLY WITHDRAWN

ment system in general consisted of (1) a sound-proofed room containing a centrifugal fan, a 3 hp motor and a sheet metal plenum chamber, (2) a second sound-proofed room lined with a 1-in. thickness of hair felt, and (3) a straight sound-proofed tunnel 76 ft in length connecting the two rooms. This tunnel contained the ducts to be tested. The walls of the rooms and tunnel were of double frame and plastered cane fiber insulation board construction. The rooms were insulated from the floor by layers of sponge rubber and hair felt, and the tunnel was supported on strips of the same materials resting on wood supports. The fan, motor and plenum chamber were likewise supported from the floor of the room, as were ducts placed in the tunnel. As far as possible, sound travel from room to room other than through the duct was prevented. The sound intensity drop from the surroundings into this system was of the order of 55 db (decibels).

Sound of various constant frequencies and intensities was generated by means of a vacuum tube oscillator feeding an amplifier and speaker. Sound

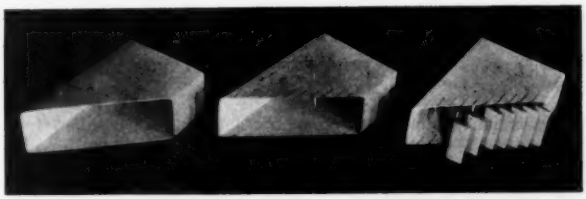

FIG. 2. FABRICATION OF 90-DEG ELL WITH ASBESTOS DIRECTING VANES

levels were measured with a sound meter reading decibels above the standard reference level of 10^{-16} w (watts) per square centimeter.

For measurements of temperature drops through various lengths of ducts, gas-heated air was fed through a centrifugal fan and plenum chamber to the ducts. Temperatures and velocities were measured at selected points along the ducts under equilibrium conditions.

FRICTION LOSSES

Most engineers will primarily be interested in a comparison of the friction losses in the asbestos and metal ducts.

To obtain this comparison, the drop in static pressure for a distance of 57 ft was obtained on 11 in. x 13 in., 5½ in. x 8½ in., 2½ in. x 11 in. rectangular and on 8-in. round asbestos and metal ducts.

Fig. 5 gives the drop in static pressure obtained on the various ducts tested. The static pressure drop corrected to a 50-ft length is plotted against the average velocity at the downstream end of the duct.

The actual coefficient of friction of smooth metal pipes has been determined by various investigators. Emswiler,[1] for instance, found that the coefficient varied from about 0.008 to 0.004 for velocities of 200 to 2800 fpm. His experiments confirmed those of other investigators in the fact that the coefficient decreases with both increase of diameter and velocity. Later investigators have found that the coefficient of friction increases with increase of viscosity; however, the effect of the viscosity of the air is small and is usually neglected. An average value of $f = 0.0054$ is quite often used.

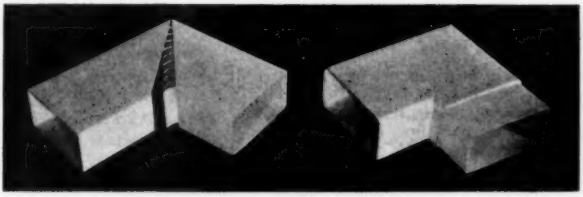

FIG. 3. ASSEMBLY OF 90-DEG ELL WITH DIRECTING VANES

[1] Coefficient of Friction of Air Flowing in Round Galvanized Iron Ducts, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916, p. 537).

The coefficient of friction for the asbestos and metal ducts was calculated from the test results by means of the relation:

$$P = \frac{fRL}{a}\,\frac{DV^2}{2g} \text{ or } h = \frac{fRL}{a}\,\frac{V^2}{2g} \tag{1}$$

in which P = pressure required to overcome friction, pounds per square foot,
a = cross-sectional area of duct, square feet.
D = density of air, pounds per cubic foot.
V = velocity, feet per second.
f = coefficient of friction.
h = height, feet of a column of air equivalent to P.
R = perimeter of round duct, feet.
L = length of duct, feet.
g = acceleration due to gravity in feet per second per second, or, in terms of inches of water, and for air at 80 F.

$$h = 0.00022\,\frac{fRL}{a}\,V^2 \text{ or } f = \frac{ha}{0.00022\,RLV^2} \tag{2}$$

Since these formulae are for round ducts, the equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity as the rectangular ducts was calculated from the relation:

$$d = 1.265\sqrt[5]{\frac{(ab)^3}{a+b}} \tag{3}$$

in which a = one side of rectangular duct.
b = other side of rectangular duct.
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity.

From formula (3), the equivalent diameters for the 11 in. x 13 in., the 5½ in. x 8½ in. and the 2½ in. x 11 in. ducts are 13.1 in., 7.49 in. and 5.5 in. respectively.

These equivalent diameters were used in calculating the coefficient of friction for the eight ducts. The static pressure drops for velocities of 1000, 2000 and 3000 fpm were taken from Fig. 5 and substituted in Equation (2). The results of the calculations are given in Table 1.

THE A.S.H.V.E. GUIDE has included for several years a chart for obtaining the friction loss in metal ducts. This chart is based on higher frictional resistances than usually occur in practice and, therefore, gives larger duct sizes so as to allow for a factor of safety for increased roughness of duct surface or for obstructions, etc.

The chart was calculated from the formula:

$$h_L = \frac{1.1\,L}{CD^{5/7}}\left(\frac{V}{4005}\right)^{13/7} \tag{4}$$

in which h_L = loss of head, inches of water.
L = length of pipe, feet.
V = velocity of air, feet per minute.
D = diameter of pipe, feet.
C = length of pipe in diameters for one head loss.
C = 60 for perfectly smooth pipe; = 55 for pipe used in planing mill exhaust systems, and 50 for heating and ventilating ducts.

Since formula (4), using a value for $C = 50$, is used so extensively in designing air conditioning ducts, it is desirable that C values be calculated

from the tests conducted on the 8 ducts. These values based on equivalent diameters of 13.1 in., 8.0 in., 7.49 in. and 5.5 in. are given in Table 1.

The static pressure drop for the same equivalent diameters as obtained from the chart in THE A.S.H.V.E. GUIDE, corrected for equivalent velocities, is included in Table 1 for comparison.

Referring to Table 1 it is seen that the values of C for the 8-in. round asbestos duct follow exactly the 1/7 power law of change in friction with change in velocity. The round metal duct is also fairly close to this relationship, and in general the rectangular asbestos and metal ducts follow closely this law for velocities of 2000 and 3000 fpm. The pressure drop is somewhat higher than this law requires for 1000 fpm in all rectangular ducts with the exception of the 2½ in. x 11 in. ducts. For these ducts the relationship is reversed and since the metal duct shows so much lower drop in relation to the other sizes, it is planned to continue the tests on the 2½ in. x 11 in. metal duct.

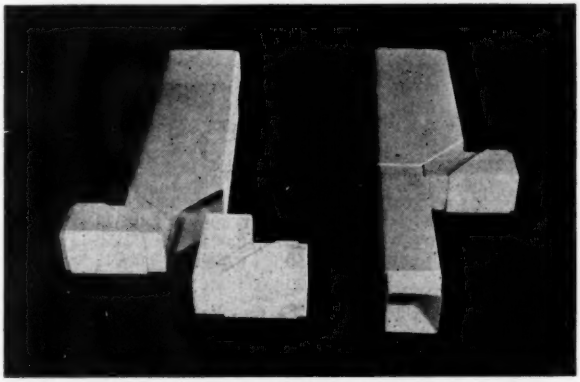

FIG. 4. ARRANGEMENT OF ASBESTOS AIR DUCT WITH TEE AND BRANCH CONNECTIONS

The variation of friction with pipe diameter in these tests did not check the 2/7 power law of change in friction with diameter of duct as closely as the check on the velocity law. However, in all cases with the exception of the 8-in. round ducts the coefficient of friction decreased with increase in equivalent diameter of duct.

An inspection of the values of C indicates that in all cases these values are considerably higher than the value of 50 recommended in THE A.S.H.V.E. GUIDE for air conditioning systems. It is noted that C for the 8-in. round metal duct is roughly 20 per cent higher than the recommended value. However, the coefficient of friction obtained for both round ducts was considerably higher than that obtained on the rectangular ducts. The higher values of C obtained on the rectangular ducts indicate that it is inaccurate to determine friction losses for rectangular ducts from the A.S.H.V.E. chart which is based on test results obtained on round ducts.

It is believed that the use of a value of 50 for C results in ducts considerably larger than is actually necessary for good engineering practice.

If one totals the drop in friction for all sizes and all velocities observed for the asbestos and the metal ducts it will be found that there is less than 1 per cent difference in pressure drop between the two types of ducts. This would indicate that there is probably no practical difference in friction between the asbestos and metal ducts.

A comparison of the friction losses through asbestos and metal fittings has

TABLE 1—RESULTS OF FRICTION TESTS ON ASBESTOS AND METAL DUCTS

Velocity, Feet per Minute	Static Pressure Drop, Inches of Water per 100 Ft			Coefficient of Friction f		Value of C	
	Asbestos	Metal		Asbestos	Metal	Asbestos	Metal
	Observed	Observed	A.S.H.V.E.				
11 in. x 13 in. Rectangular Ducts							
1000	0.098	0.098	0.165	0.00391	0.00391	84.5	84.5
2000	0.310	0.310	0.600	0.00308	0.00308	97.0	97.0
3000	0.645	0.670	1.270	0.00286	0.00297	98.3	94.8
8 in. Round Ducts							
1000	0.200	0.235	0.280	0.00543	0.00638	70.0	59.5
2000	0.725	0.825	1.020	0.00492	0.00560	70.0	61.7
3000	1.545	1.735	2.150	0.00466	0.00522	70.0	62.5
5½ in. x 8½ in. Rectangular Ducts							
1000	0.200	0.195	0.340	0.00420	0.00443	85.0	87.2
2000	0.665	0.595	1.270	0.00377	0.00339	92.5	103.5
3000	1.395	1.300	2.600	0.00353	0.00328	93.5	100.0
2½ in. x 11 in. Rectangular Ducts							
1000	0.341	0.267	0.600	0.00474	0.00371	93.2	118.5
2000	1.175	1.032	2.500	0.00410	0.00360	89.8	102.0

been made on several sizes of 90-deg elbows, and it is assumed that data on these fittings are applicable, for purposes of comparison, to other fittings.

Fig. 6 shows the experimental values of plenum chamber static pressure necessary to force air, at the velocities given, through 42 ft of 5½ in. x 8½ in. asbestos duct equipped (1), with a metal 90-deg broadway sweep turn of radius ratio 1.5 and (2), with a 90-deg asbestos square turn equipped with asbestos turning vanes. Within the limits of experimental error, the loss is the same for both turns, and under experimental conditions, the loss through each was about 22 per cent of the velocity head for all velocities.

Published values for the friction losses through metal elbows of square

cross-section and radius ratio equal to 1.5 vary from 9 per cent to 40 per cent of the velocity head,[2] depending upon the conditions. The values plotted in Fig. 6 were obtained with 24 ft of duct between the plenum chamber and the turn, and the turn was followed by 18 ft of additional duct.

With the metal turn under test, the velocity distribution was not uniform across the end of the 18-ft leg after a distance corresponding to nearly 30

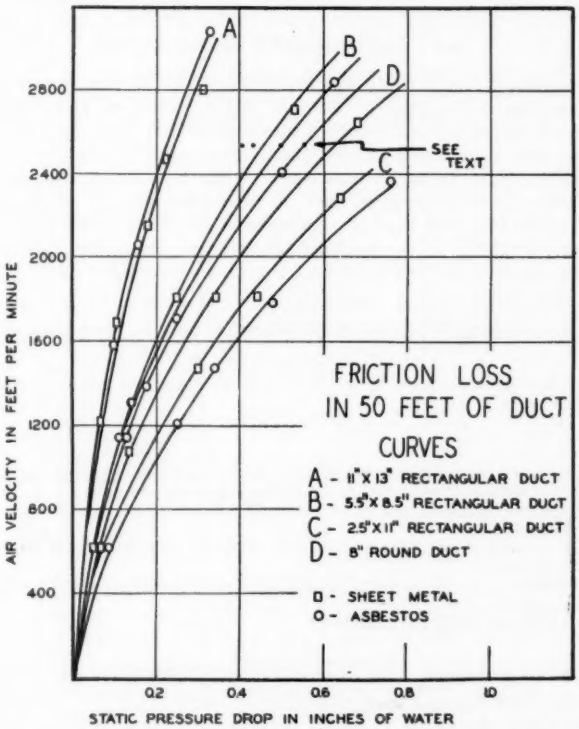

FIG. 5. FRICTION LOSS IN 50 FT OF ASBESTOS AND METAL DUCTS

diameters of duct. Any evaluation of the losses in turns, or in ducts, for that matter, should be accompanied by the conditions of test. Air straighteners are commonly used in tests of this nature. Straighteners are not commonly used in practice, and tests made therewith should not be used to compute losses in practice. The test shown in Fig. 6 was merely comparative, and no attempt was made to measure other than plenum chamber pressures and average duct velocities. The entrance loss to the duct was known to be very high. Values

[2] Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, August, September, 1936).

obtained for the loss in identical turns varied from 85 per cent of the velocity head to 17 per cent, depending entirely on conditions.

Points indicated on Fig. 5 by the arrow show observed values of the friction loss in 5½ in. x 8½ in. asbestos ducts; the apparent friction loss per unit length decreased markedly with increase in length of tested duct. No apology is offered for the method of test, as the systems used simulate practical installations, and all curves in Fig. 5 were obtained under comparable conditions.

HEAT TRANSMISSION CHARACTERISTICS

Two conductivity tests were run on ½-in. thick asbestos duct samples and two conductivity tests were run on 1-in. thick asbestos duct samples. Since there was no appreciable difference in the conductivity (per inch of thickness) of the 1-in. and ½-in. thick samples the average conductivity of the four tests only is given. The thermal *conductance* of the 1-in. thick duct is of course one-half that of the ½-in. thick duct. The conductivity as measured in these tests was 0.622 Btu at a mean temperature of 50 F and 0.707 Btu at a mean temperature of 150 F. The conductivity at mean temperatures between these two values is a straight line function of mean temperature.

The thermal transmission coefficient U for various temperature conditions, etc., can be calculated from the standard formula,

$$U = \frac{1}{\frac{1}{f_i} + \frac{L}{K} + \frac{1}{f_o}} \qquad (5)$$

where

U = Btu per square foot per hour per degree Fahrenheit difference in temperature between the air inside and the air outside the duct.
f_i = film conductance inside the duct, Btu per hour per square foot per degree Fahrenheit.
f_o = film conductance outside the duct, Btu per hour per square foot per degree Fahrenheit.
K = conductivity of the duct, Btu per hour per square foot per degree Fahrenheit per inch.
L = thickness of the duct, inches.

It will be found that f_i will vary considerably with the velocity and the size of the duct. THE A.S.H.V.E. GUIDE gives a value of $f_i = 6$ for outside walls and for a wind velocity of 15 mph. Various investigators show lower values than this for ducts 6 in. in diameter or over. For the purpose of making approximate calculations an average value of $f_i = 4$ has been used in this paper.

The outside surface conductance f_o will vary considerably with the size of the duct for small ducts and for metal ducts it will vary considerably with the emissivity p of the duct. Values of f_o for various conditions can be calculated from data given by Heilman [3] and others. Previous tests were conducted on galvanized metal to determine the emissivity of bright and tarnished material. Very bright galvanized metal had an emissivity of 0.14, while badly tarnished

[3] Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July, 1936, p. 782).

metal had an emissivity of 0.53. Horizontal metal ducts will ordinarily be covered with a layer of dust on the top surfaces, so that part of the duct will have a very high emissivity value. Some metal ducts are covered with asbestos paper. In this case the emissivity is 0.93.

Tests were conducted on various sizes of ½-in. thick asbestos ducts and on an 11 in. x 13 in. metal duct insulated with ½-in. thickness of cellular laminated asbestos paper insulation to determine the drop in temperature in 30 ft of duct. These tests were run with velocities of approximately 650 to 1750 fpm and with initial air temperatures of approximately 160-230 F.

THE A.S.H.V.E. GUIDE 1937, page 176, gives a formula for calculating the temperature drop in ducts. For low velocities and long ducts of small cross-section a somewhat more accurate formula may be used as follows:

$$T = \frac{T_i - T_o}{e^{\left(\frac{UPL}{60\,Advc}\right)}} + T_o \qquad (6)$$

where

T = final temperature of air in duct, degrees Fahrenheit.
Ti = initial temperature of air in duct, degrees Fahrenheit.
T_o = outside air temperature, degrees Fahrenheit.
U = thermal transmittance of duct, Btu per hour per square foot per degree Fahrenheit.
P = perimeter of duct, feet.
L = length of duct, feet.
A = area of duct, square feet.
d = density of air, pounds per cubic foot.
v = velocity of air, feet per minute.
c = specific heat of air, Btu per pound per degree Fahrenheit.
e = Naperian base of logarithms = 2.718.

Equation (6) can also be used in calculating the thermal transmittance U of a duct when the drop in temperature through a given length of duct is known or measured. The thermal transmittance for the various sizes of asbestos ducts on which temperature drop determinations were made and also for the insulated metal duct have been calculated from equation (6) and the observed temperature drops. The average transmittances for three sizes of asbestos ducts with an average internal temperature of 166 F and an outside temperature of 85 F and the transmittance for the metal duct insulated with ½-in. thick cellular laminated asbestos paper for the same temperature conditions are given in Table 2.

Thermal transmittance values have also been calculated for various types of ducts from equation (5) and the known values of K as determined for asbestos ducts and cellular laminated asbestos paper insulation. In calculating the transmittance values for the bright, the tarnished, and the asbestos paper-covered metal ducts, the emissivity values used were 0.14, 0.53 and 0.93, respectively. In these calculations an average air temperature of 166 F in the duct and an outside air temperature of 85 F were used. A value of 4 was used for f_1. The results of these calculations are as given in Table 2.

TABLE 2—THERMAL TRANSMISSION COEFFICIENTS FOR METAL AND ASBESTOS DUCTS

DUCTS	BTU PER HOUR PER SQUARE FOOT PER DEGREE FAHRENHEIT, U
½ In. Thick Asbestos, Calculated	0.662
½ In. Thick Asbestos, Observed	0.668
Bright Metal	0.832
Tarnished Metal	1.11
Metal, Covered with Asbestos Paper	1.33
Metal, Insulated with ½ In. Thick Cellular Laminated Asbestos Paper, Calculated	0.482
Metal, Insulated with ½ In. Thick Cellular Laminated Asbestos Paper, Observed	0.523

The heat transfer from metal and asbestos ducts is lower for air cooling installations owing to the lower temperatures and to lower temperature differences.

The heat transmission in Btu per square foot per hour for various types of ducts as calculated from equation (5) is given in Table 3. These calculations were based on an average temperature of 60 F in the duct with an outside air

TABLE 3—HEAT TRANSMISSION VALUES FOR METAL AND ASBESTOS DUCTS

DUCTS	HEAT TRANSMISSION, BTU PER SQUARE FOOT PER HOUR
½ In. Thick Asbestos	12.1
Tarnished Metal	17.5
Metal, Covered with Asbestos Paper	23
Metal, Covered with ½ In. Thick Cellular Laminated Asbestos Paper	8.8

temperature of 80 F. The emissivity values and the film conductances which were used in the previous calculations were also used in these calculations.

The drops in temperature through 50 ft of various sizes of ½-in. thick asbestos ducts, tarnished metal ducts, and metal ducts insulated with ½-in. thick cellular laminated asbestos paper have been calculated by means of equation (6) using coefficient of transmission values of 0.662, 1.11 and 0.482

TABLE 4—CALCULATED TEMPERATURE DROP THROUGH 50 FT OF DUCT INITIALLY AT 175 F, OUTSIDE AIR TEMPERATURE, 70 F

DUCT SIZE	VELOCITY 500 FPM			VELOCITY 1000 FPM		
	Asbestos Duct	Tarnished Metal	Insulated Metal	Asbestos Duct	Tarnished Metal	Insulated Metal
2½ in. x 11½ in.	59.5	79.1	48.0	36.0	53.0	27.6
5½ in. x 11½ in.	38.8	66.2	38.0	21.7	34.2	16.2
11½ in. x 11½ in.	27.1	41.5	20.6	14.7	23.3	10.7
11½ in. x 23½ in.	21.0	32.7	15.7	11.5	17.9	8.1
23½ in. x 23½ in.	14.4	22.8	10.8	7.4	12.3	5.5

respectively. In these calculations an outside air temperature of 70 F was assumed. The calculated values are given in Table 4.

SOUND CHARACTERISTICS

Various tests of a practical nature were made on the behavior of galvanized sheet metal ducts and asbestos ducts with respect to sound transmission.

In general, intensities of sound generated in a plenum chamber by means of an oscillator of variable frequency feeding a speaker at variable intensities, or by means of a fan, blowing air through the system, were measured at both ends of ducts of various sizes but of fixed length at 76 ft. The system was sufficiently well insulated that extraneous noise effects as well as transmission

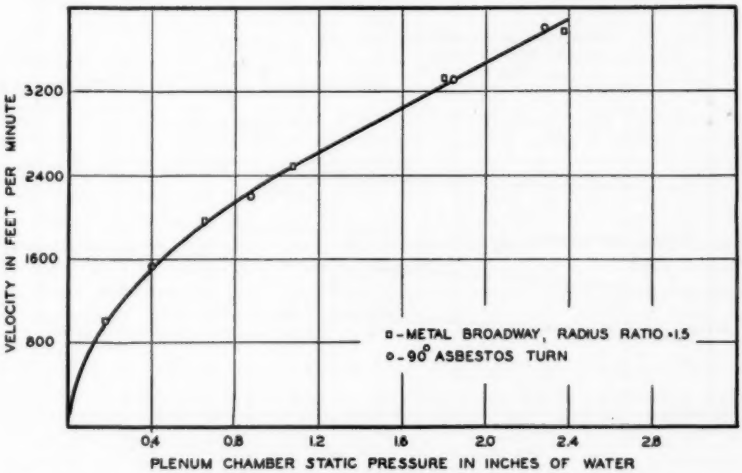

FIG. 6. PLENUM CHAMBER STATIC PRESSURE TO FORCE AIR THROUGH 42 FT OF 5½ IN. X 8½ IN. DUCT INCLUDING 90-DEG TURN

of noise from the sources used other than through the ducts under test affected the measured levels to an unmeasurable degree.

The oscillator frequencies available were limited to 100, 250, 500, 1000 and 2000 cycles per second. Obtaining definite values of the actual transmission of sound through ducts under these conditions is very difficult, as has been indicated in the literature on sound measurement. The chief difficulty in measurement of levels lies in the variable sound intensity pattern, which shifts with each frequency used and which, also, prevents exact exploration of duct openings other than mechanically, as the observer's body has marked effect on measured intensity levels.

The data obtained are of such nature that only indications may be mentioned. As is to be expected, asbestos ducts transmit less sound than metal ducts, the difference increasing with frequency.

Due presumably to the large number of component frequencies of the sound

developed (eliminating a fixed intensity pattern in the rooms), the noise levels caused by the fan and the air it fed through the duct were measured with relative ease and the values obtained were reasonably reproduced.

In Fig. 7 are shown curves comparing the sound intensities in the room at the end of the ducts indicated and the intensity drops through the ducts at the given air velocities. The measured sound level in the plenum chamber was about 90 db above the reference level of 10^{-16} w per square centimeter. The level in the plenum chamber at duct air velocities of 3500 fpm was about 2 db higher than at a velocity of 500 fpm. That this difference is no greater is probably due to the fact that the air velocity was varied by means of a damper at the fan inlet; the fan noise should be about the same for widely

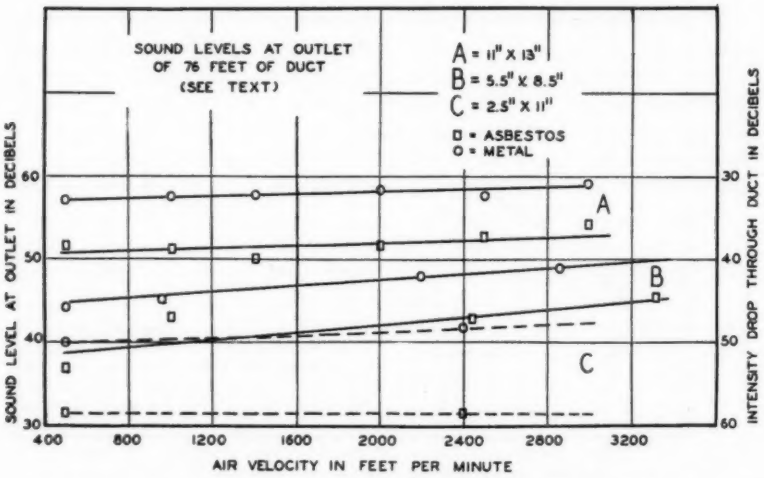

FIG. 7. COMPARISON OF SOUND INTENSITY DROPS THROUGH 76 FT OF METAL AND ASBESTOS DUCTS

different velocities. In practice, a smaller fan would be used for lower velocities; for this reason the plenum chamber sound levels at lower velocities probably would be lower than those given. All data in Fig. 7 are referred to a plenum chamber noise level of 90 db.

The drops in sound intensity through asbestos ducts of all sizes are higher than those through the corresponding metal ducts, and the improvement is substantially constant up to velocities of 3500 fpm. At a velocity of 1000 fpm, the sound levels in the room at the exhaust end of the ducts were lower with asbestos ducts by the following amounts: 2½ in. x 11 in. ducts, 8.8 db; 5½ in. x 8½ in. ducts, 5.7 db; 11 in. x 13 in. ducts, 6.0 db.

It should be noted that the levels given in Fig. 7 are higher than ordinarily obtained; levels were kept high to facilitate accurate measurement under variable conditions, and the levels were obtained close to the duct outlets. The ducts tested were straight, with no fittings, turns or other sound absorbing

equipment which would have made the levels lower in an actual installation. Plotted levels at this point are the averages along fixed points in the room, and all data were obtained under standardized conditions. The effect of the standing wave patterns and of the observer's body are negligible when making observations on sound of this nature.

For the intensity levels given in Fig. 7, the average apparent loudness of the noise issuing from the asbestos ducts is as follows:[4]

VELOCITY, 1000 FPM

2½ in. x 11 in. asbestos duct, loudness approximately 40 per cent that of metal.
5½ in. x 8½ in. asbestos duct, loudness approximately 60 per cent that of metal.
11 in. x 13 in. asbestos duct, loudness approximately 65 per cent that of metal.

It has been noted that the sound issuing from the metal ducts under test contained equipment noise to a variable extent. With the asbestos ducts equipment noise apparently is absent, under comparable conditions.

The data of Fig 7 also indicate that air velocities in asbestos ducts may be increased greatly over present customary velocities without exceeding the present sound intensities at the duct outlets.

Data obtained using fixed frequencies check within the limits of error the data obtained with air. However, the data at low frequencies indicate that for small ducts, as tested here, the entrance and exit drops in sound level predicted by the theory of sound transmission through conduits of cross-section much less than one wave length of sound are of considerably more importance, and drops obtained at low frequencies are higher than would be expected from the data obtained at higher frequencies.

The sound level drop through a 5½ in. x 8½ in., 90 deg asbestos turn, equipped with asbestos turning vanes, was compared with that through a metal sweep turn of the same cross-section and of radius ratio 1.5, both with sound of fixed frequency and with air. The asbestos turn decreased sound of frequencies 250, 500 and 1000 cycles per second approximately 7, 11 and 14 db, respectively, more than the metal turn, which had virtually no effect. With air the differences were approximately 4 and 7 db for velocities of 500 and 1000 fpm. There was no measurable difference between the two turns at 2000 and 3000 fpm. In a system containing several asbestos fittings, a much lower exit sound level may be expected at ordinary velocities than in systems that contain only metal sweep turns.

Sound is picked up by asbestos ducts less readily than by bare metal ducts, the effect increasing considerably with frequency of the sound and thickness of the duct. The sound picked up by a duct, independently of that by conduction, should be approximately inversely proportional to the weight per square foot of area.[5] The value of a duct of this nature lies in its ability to absorb picked up sound. As is to be expected, there is no sounding board effect on asbestos duct with changes in temperature, with opening of dampers, etc., and the sound of dampers and equipment noises are considerably diminished over

[4] American Tentative Standards for Noise Measurement (*American Standards Association*, Feb. 17, 1936).
[5] Acoustics and Architecture, by Paul E. Sabine (McGraw-Hill, 1935, p. 279).
Acoustics, by Geo. W. Stewart and R. B. Lindsay (D. Van Nostrand and Co., 1930, p. 130).

those connected to metal ducts. There is no measurable sound conduction through the material itself. The sound intensity measured at 3 ft from the side of a metal duct at the end of which a damper was opened and closed, starting and stopping air flow, was about 90 db. This is probably higher than is ordinarily the case with dampers.

CONCLUSIONS

The results of this preliminary investigation indicate that the asbestos ducts have practically the same coefficient of friction as metal ducts. The tests also indicate that the Friction Chart given in THE A.S.H.V.E. GUIDE requires considerably larger ducts than is necessary for good engineering practice.

Conductivity data on the asbestos duct are given. Also, calculated and measured temperature drops for various ducts are included.

The tests indicate that present usual air velocities can be increased many times without undue noise in the asbestos ducts. For all air velocities tested, the asbestos ducts were superior to metal ducts in sound attenuation.

DISCUSSION

R. R. GANNON[6] (WRITTEN): We have carefully read this paper and, in order to be brief, have compiled our discussion in the form of questions. The answers to these questions we hope will afford profitable discussion.

1. Since all data appear to be based on clean surfaces inside asbestos ducts, what effect does accumulated dirt have on the film conductance (f_1) inside the duct and the emissivity?
2. What effect would dirt inside sheet metal ducts have on their film conductance and emissivity?
3. If dirt on the surface of ducts changes their emissivity, of what value are the data published? Is it possible and practical to publish data based on ducts in a condition in which they are found in practice?
4. Is it practical and advisable, from an acoustical standpoint, to line the asbestos duct for a short distance from the fan (or source of noise) with a material having better sound absorbing qualities?
5. How does the cost of an installation using asbestos duct work compare with one using sheet metal and sheet metal covered with insulation referred to in your paper? What are the compensating factors?
6. Since one of the greatest contributors to air conditioning troubles is the well known draft—how can an installation be made, using asbestos ducts with higher velocities than are now used, without increasing this trouble?

We feel that this development is leading the way to a long felt need for a more satisfactory means for conveying air from one place to another. The authors of this paper are to be commended for this development.

W. W. PEARSE:[7] I would like to ask the speaker if the connecting links of the duct run entirely through the duct or are they merely short connecting links between the insulated exteriors?

R. H. HEILMAN: It runs entirely through the duct. In other words, there is an inner core and an outer core. The inner core should be telescoped ordinarily about

[6] Russell R. Gannon Co., Cincinnati, Ohio.
[7] Toronto, Ont., Canada.

halfway through the section; that is, they are 3-ft sections and that means your joint comes in 18 in. from the outer joint of the jacket. For the air to get out it is necessary for it to travel 18 in. back through between the inner core and the outer jacket and then out. We find that there is practically no leakage loss in the duct. We do not get that with the metal duct because you cannot get it as tight unless you use some kind of a seal.

I did not get all the questions down but the first one was the effect on the inner film conductance of dust on the inner surface of the duct. It would not affect the conductance unless it was very thick and even then it would not affect the film conductance on the inner asbestos duct because asbestos has an emissivity of about 0.93. Dust would be slightly higher than that, approximately 0.97, which, of course, would be very immaterial, but it would have an effect on the film conductance for the metal duct. In that case the rate of heat transfer would be greater through the metal duct than it would have been if you did not have a film there, for the simple reason that you would not have the reflecting surface; it would decrease the effect of the reflecting surface.

W. H. CARRIER: You are speaking of the inside now?

MR. HEILMAN: Inside.

MR. CARRIER: I do not know where you have that emissivity on the inside.

MR. HEILMAN: A bright metal duct.

MR. CARRIER: With what?

MR. HEILMAN: Air that has a certain amount of water vapor in it. In other words, there are certain gases that do not obey the ordinary laws of radiation, and they will radiate heat, that is they will have a certain effective radiant energy under certain wave lengths. This has been discovered and worked out by several prominent investigators, and they found, depending of course on the depth of the path of the gas, that you can get emissivity values for a gas as high as probably 95 per cent. Ordinary air is absolutely transparent to thermal radiation, but if you get a certain amount of water vapor in it, then it is not absolutely transparent. In that case you will have some effect. Of course, it will be slight.

With reference to the question, If dirt on the surface of ducts changes their emissivity, of what value are the data published? I want to say that I gave values for a bright metal duct. The convection values will not change; it is just the emissivity values that will change, and, of course, the film coefficient is tied up in both radiation and convection. The bright metal duct, when you get it from a sheet metal man before it is erected, may have an emissivity value as low as 0.14. I think everybody here understands what that means. The emissivity of a perfect black body is one. Everything else is less than one, so the radiation is directly proportional to the emissivity value, so if you have a unit loss of 100 Btu with an emissivity value of one, with an emissivity value of 0.14 you would have only a loss of 14 Btu. If you have a very tarnished metal the emissivity value will go as high possibly as 0.53. That was the highest value I obtained on an artificially tarnished metal surface.

Asbestos paper has an emissivity of 0.93, so it will not change. If you have dust collecting on the outer surface of your metal duct, as you probably would especially on a horizontal portion, then the emissivity value of the metal duct will be about 0.93 to 0.95, so I think that will answer that part of the question.

It is practical from an acoustical standpoint to line the asbestos duct for a short distance, but I do not believe it was brought out in the paper; I did not have time to mention it, although it is mentioned in the body of the paper, that one turn cuts down the sound intensity practically the same amount as a 76-ft length of duct. In other words, if you have a number of turns, as you would have in an ordinary installation, you are going to cut down the intensity to a very high percentage; it probably would not be necessary to have internal acoustical material.

In regard to the cost, that is pretty hard to answer exactly, but we think the

erected cost will be approximately midway between a bare metal duct and an insulated metal duct, that is, installed cost.

It has been asked how an installation can be made using asbestos duct with higher velocities. I am not so familiar with that phase of the subject, but it is my impression that you can widen out your diffusers and cut down your velocity at the outlet. You can increase the velocity in the asbestos duct itself. We ran tests up to 3500 ft. You could run these up to 5000 or 10,000 ft and it would not affect the duct and it does not apparently increase the noise.

In other words, at 10,000 ft there would be no more noise in the duct than at 2000 ft in a metal duct.

E. R. SNAVELY: What is the effect of condensate on this duct that goes through an atmosphere which is so cold that you get some condensation?

MR. HEILMAN: We expect to be able to have the duct waterproofed. Materials have been made although we have not tried them on this particular duct. Treated asbestos is soaked in water for 24 hours and it has to have enough strength so that after soaking for 24 hours in water a 1-in. square plunger, having a weight of 60 lb on it, will not cause breakage of the surface. It is possible to make asbestos paper very strong and rigid and still have very good insulating value.

E. H. LLOYD: Under conditions of intermittent operation, do you observe any lag in the temperature of air at the end of a long run of duct work? In other words, I assume these temperature drops in Table 4 are based on steady operation. Now, what would you experience with this asbestos duct as compared to a metal duct, or a metal duct insulated, in the way of temperature lag, for instance under 10- or 15-min operation periods?

MR. HEILMAN: With the insulated metal and the asbestos, there should be no difference because they are both insulated. The only difference that you get in the lag is, of course, in heat stored in the insulation, and so there would be no difference between the asbestos duct and the insulated duct provided they both had the same specific heat and the same density. Of course, there would be some difference in that. With the uninsulated metal duct you will have less lag.

PROF. C. P. YAGLOU: I should like to ask the author if there would be any odor imparted to humid air passing through the asbestos duct. Would there be any odor in the rooms?

MR. HEILMAN: If the asbestos duct is waterproofed, there should be no odor, and there really shouldn't be any odor if it isn't. If the inside would get so contaminated with the gases that are going through for a continuous length of time a certain amount of that odor might be deposited in the duct and would naturally come out, but I think the possibility of that would be rather slight, although I could not say absolutely.

No. 1083

EFFECTS OF ARTIFICIAL LIGHTING ON AIR CONDITIONING

By WALTER STURROCK *(*NON-MEMBER*), CLEVELAND, OHIO

THE air conditioning of a building or any part of it for the summer months involves several problems, one of which is that of maintaining comfortable air temperatures. The discussion in this paper refers to the effect of artificial lighting on the temperature rise which must be considered in designing the cooling system. It is needless to state that natural light from the sun and sky coming through windows carries with it a considerable amount of radiant energy much of which becomes sensible heat when it strikes an absorbing surface. Likewise, artificial light sources emit heat, most of which is in the form of radiant energy, but becomes sensible heat as soon as it strikes an absorbing surface.

Reproduction of the so-called cold light, such as that made by the firefly, is known to be possible by mixing a luminol-caustic soda solution with a hydrogen peroxide-potassium ferricyanide solution. The cost, however, of producing cold light by this method is extremely high. It is more than a million times as great as that with modern incandescent lamps, and therefore needs no further consideration here. All practical light sources generate heat as well as light, and there is a definite heat equivalent for each watthour of energy consumed, which amounts to 3.415 Btu. While this heat equivalent per watthour for electric lamps is a constant value, it is interesting to note that due to the many researches, principally during the past quarter century, the light units generated by each watt have gradually increased. For example, the 200-w (watt) gas-filled lamp, which has been on the market more than 20 years, appears today about the same as when first made available, but its light output has changed as follows: 20 years ago 1 kw of lighting load using five 200-w lamps produced about 14,000 lumens of light; 15 years ago it produced 15,100 lumens; 10 years ago 16,200; 5 years ago 16,900, and today it has increased to 18,200. This last increase in light output was made possible by the high efficiency 750-hour lamp which became practical by recent price reductions. The increases in lumens per watt over a 20-year period are shown graphically in Fig. 1, and also the reduction in lamp prices, both of which are important factors as far as the economics of artificial lighting is concerned.

The levels of illumination (footcandles) obtained from artificial lighting, in

* Illuminating Engr., Nela Park Engr. Dept., General Electric Co.

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1938.

general, are so low in comparison to the thousands of footcandles found out-of-doors that the public has not only always been eager to accept increases in the efficiency of producing light in order to provide better illumination, but has in many instances demanded double or perhaps triple the connected wattage load used 10 to 15 years ago. During all these advances there is, of course, no change in the fact that each watt of lighting load continues to generate 3.415 Btu per hour.

HEATING EFFECT FROM LIGHTING INSTALLATIONS

From the growing interest in adequate lighting in offices, stores, factories, and other work places, the illumination levels for the future will unquestionably be higher than those of the past. These increased levels will in many cases be

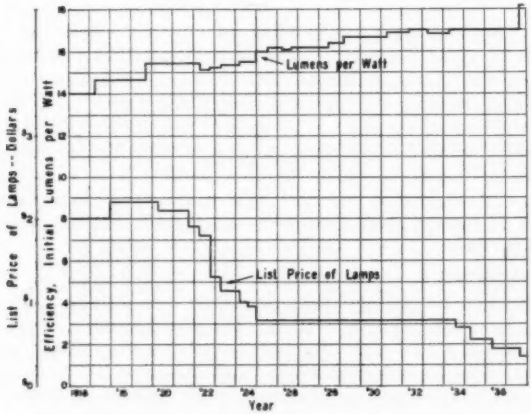

FIG. 1. EFFICIENCY AND COST CURVES FOR 200-W INCANDESCENT LAMPS

obtained by simply using higher wattage lamps. In other cases, and particularly where air conditioning is to be used, it will be desirable to revamp the entire lighting system or in new buildings to give special attention to the lighting in order to obtain a high utilization of light with its corresponding minimum wattage requirement. Some of the considerations along this line of development are discussed in this paper.

Higher Utilization of Light

The total wattage required to provide a specified level of illumination for an interior depends to a great extent on the percentage of the light generated by the lamp (utilization coefficient) which actually reaches the working area. This utilization of light depends not only upon the ability of the sidewalls and ceiling to redirect light but also upon the total light output of the reflector or diffusing globe in which the lamp is placed. Recent developments in higher reflection factors for paints and in the light reflection properties of aluminum

are of special interest. A new finish, known as Alzak, has not only increased the light output of aluminum reflectors but also has provided a surface which resists abrasion and when dirty can be readily cleaned to restore its initial efficiency. A recent lighting installation for a small office, shown in Fig. 2, was designed so that small indirect aluminum reflectors directed the light toward a special ceiling which was also of aluminum. In this case, the utilization of light was 32 per cent as compared with 24 per cent which would have been obtained from a conventional lighting system. In other words, in order to get

FIG. 2. PARABOLIC SECTIONAL DESIGNED ALUMINUM CEILING IN OFFICE PROVIDES 33 PER CENT HIGHER UTILIZATION OF LIGHT THAN ORDINARY INDIRECT LIGHTING

an equal level of illumination from a conventional system the wattage load would have been 1⅓ times as great.

A large public utility located in Detroit has recently completed the lighting design for a new office building. A feature of particular interest is the use of ceiling coffers for lighting units. They are about 4 ft square and 15 in. deep and are designed to utilize the space between the large air ducts in the ceiling running lengthwise of the building. By using a silvered bowl lamp in each of these coffers the utilization of light is approximately 42 per cent, which is exceptionally good. An ordinary indirect system if installed would have had a utilization of not over 28 per cent, which would have required 50 per cent more

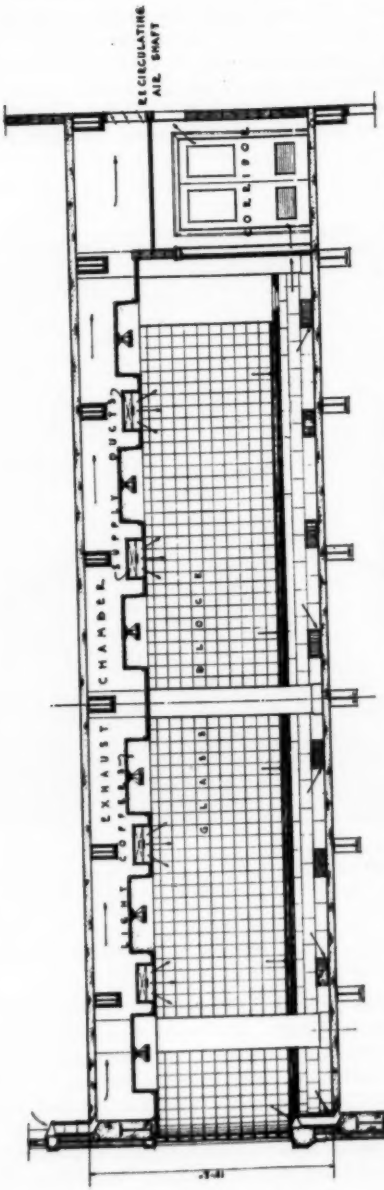

FIG. 3. AIR DUCTS FOR AIR CONDITIONING AND LARGE COFFERS EMPLOYING SILVERED BOWL LAMPS FOR LIGHTING INSTALLED IN THE FALSE CEILING

watts to provide the 35 footcandles in service obtained from the adopted system. Fig. 3 shows a cross-section of a typical floor. Other examples of well-designed lighting systems could be discussed, but suffice it to say that more thought should be given to this phase of the problem especially for new or remodeled buildings where it is definitely known that an air cooling system will be installed.

Minimizing Heat by Reduction of Light

The quantity of heat accompanying light can always be reduced by making a corresponding reduction in the footcandles of illumination. In the case of natural light coming through the windows, awnings can be installed to intercept the radiant energy but likewise they intercept a high proportion of the light and are often objectionable for this reason. In artificial lighting, smaller sized lamps will reduce the heat as well as light, but this is objectionable from the standpoint of the footcandles required. If lower wattage lamps are used to reduce the heat during the summer months, when air cooling is desired, then in the same manner as awnings are stored away in the winter the lower wattage lamps should be replaced early in the fall by lamps of the proper size to provide adequate illumination. A replacement of lamps at different seasons of the year is not, however, in general a practical solution to the problem. For buildings having considerable window area for natural light a better solution is to arrange the wiring system so that the lamps near the windows can be turned off at times when daylight is sufficient in this area. During the summer months the turning off of lamps near the window in these buildings would not be serious. On the other hand, during the winter months artificial lighting is needed over the entire area to supplement daylight, particularly for the early forenoon and late afternoon hours.

Psychological Effects of Lighting

The yellowish appearance of some artificial light sources makes a room appear warmer. On the other hand, artificial light sources producing a whiter light make a room psychologically cooler even though lamps of a higher wattage may be used.

With the adoption of higher levels of illumination, particularly in smaller rooms, the color of the vertical surfaces in the line of vision has become a very important psychological factor. From this point of view the cooler colors in side wall finishes are desirable. In selecting these cooler colors, it is not necessary to choose a side wall finish of extremely low reflection factor, which would not only be depressing but would also create an undesirable contrast. In general, side wall finishes having a reflection factor of 45 to 55 per cent are quite satisfactory. Colors such as the light greens, light blues and various mixtures of them come within this range. They have been satisfactorily used in many interiors where 30 or more footcandles of illumination were installed.

It is believed that psychological factors deserve careful consideration in connection with the advancement of air conditioning along with higher levels of general lighting.

STANDARDS OF ILLUMINATION

There is no constant ratio showing the relation between the level of illumination and the wattage consumed for lighting. Larger lamps are more efficient

TABLE 1—A GROUP OF INTERIORS WITH RECOMMENDED MINIMUM STANDARDS OF ILLUMINATION AND THE APPROXIMATE LIGHTING REQUIREMENT IN WATTS PER SQUARE FOOT OF FLOOR AREA

	RECOMMENDED FOOT-CANDLES	ELECTRICAL DEMAND WATTS PER SQUARE FOOT
Auditorium	5	1.5
Automobile Show Room	20	4.0
Banks		
Lobby	10	2.5
Counters (75 w per running foot)	50–100	
Barber Shop and Beauty Parlor	20	4.0
Bowling		
Alley Runway and Seats	10	2.5
Pins (300 w per set)	30–50	
Billiards		
General	10	2.5
Tables (450 w per table)	30–50	
Club and Lodge Rooms		
Lounge and Reading	20	4.0
Auditorium	5	1.5
Court Rooms	10	2.5
Dance Halls (with no color or spectacular effects)	5	1.5
Drafting Rooms	30	7.0
Hotels		
Lobby (with no provision for convention exhibits)	10	2.5
Dining Room	5	2.5
Kitchen	10	2.5
Bedrooms (including allowance for table and bed lamps)	10	3.0
Corridors (10 w per running foot)	2	
Writing Room (including allowance for portable lamps)	20	4.0
Library		
Reading Rooms (including allowance for table lamps)	20	6.0
Stack Room (12 w per running foot of facing stacks)	10	
Moving Picture Theaters (with no color or spectacular effects)		
During Intermission	5	1.5
During Pictures	0.1	0.5
Museum		
General	10	2.5
Special Exhibits (an average of 2.5 w per square foot)	50–100	
Office Buildings		
Private and General Office		
No close work	10	3.0
Close work	20	5.0
Conference Rooms	10	3.0
Filing and Mail Sorting	20	5.0
Reception Rooms	10	3.0
Corridors and Stairways	5	1.5
Post Office		
Lobby	10	3.0
Sorting, Mailing, etc	20	4.0
Office, Private and General	20	5.0
File Room and Vault	10	2.5
Corridors and Stairways	5	1.5
Professional Offices		
Waiting Rooms	10	2.5
Consultation Rooms	20	5.0
Operating Rooms	20	5.0
Dental Chairs (600 w per chair)	30–50	

TABLE 1—(*Continued*)

	RECOMMENDED FOOT-CANDLES	ELECTRICAL DEMAND WATTS PER SQUARE FOOT
Restaurants, Lunch Rooms and Cafeterias		
Dining Area	10	2.5
Food Displays (100 w per running foot of counter)	30–50	
Stores		
Main business districts—large cities	20	5.0
Neighborhood stores—large cities and all stores—small cities and towns	15	3.5
Show Cases		
40 w per running foot	50–100	
Show Windows		
Brightly Lighted Districts, large cities 300 w per running foot of glass	200	
Secondary business districts, large cities 150 w per running foot of glass	100	
Neighborhood stores, large cities and for small cities and towns—75 w per running foot of glass	50	
Lighting to reduce daylight reflections—500 w per running foot of glass	400	
Theaters		
Auditoriums (with no color or spectacular effects)	5	1.5
Foyer	10	2.5
Lobby	15	3.5

in light production than smaller ones. Furthermore, the light output and distribution efficiency of certain lighting units give them an outstanding advantage over other lighting units when installed for a particular application. Moreover, in some cases, double the wattage is required as in others in order to provide the same level of illumination. Therefore, the best procedure for determining the actual wattage consumed by the lighting system is to survey the installation and make a summation of the lamp sizes. In new or remodeled buildings the summation of lamp wattages should be obtained from approved and accepted lighting specifications. In any case it will be helpful to know the minimum footcandle values and the corresponding approximate watts required for modern lighting levels in various interiors in which air conditioning may be desired. Table 1 shows a list of interiors with the approximate lighting loads given either in watts per square foot of floor area, or in other terms as indicated. The recommended footcandles represent the order of magnitude rather than exact levels of illumination.

ENERGY ANALYSIS FROM INCANDESCENT LAMPS

When a gas-filled incandescent lamp of the 300 to 500-w size is operated at normal voltage in still air its energy distribution is about as given:[1]

Radiation in the visible spectrum ..... 11 per cent
Heat as invisible radiation in the infra-red region ..... 70 per cent

[1] The Tungsten Lamp, by W. E. Forsythe and E. M. Watson. (*Journal of Franklin Institute*, Vol. 213, No. 6, June, 1932, p. 623).

Heat which is conducted away from the filament through the filament supports and leads	3 per cent
Heat dissipated by gas convection and conduction	8 per cent
Heat radiated by the bulb	8 per cent

Hence from a clear-bulb, gas-filled lamp about 90 per cent of the total energy is in radiant form; *i.e.*, all of it except that dissipated by the filament supports and leads, and by the gas. In burning a bare lamp in an interior its radiant energy will not be effective in raising the temperature in that interior until it has been intercepted by an absorbing surface which in turn will dissipate the heat by convection. The radiant energy will, however, increase the feeling of warmth to the human body by its radiant effect. Practically all surfaces such as building materials, room furnishings, and lighting equipment absorb a part of the wave energy reaching them from a burning lamp. As far as the lighting equipment is concerned the amount of radiant energy intercepted by it will depend upon its type.

One of the most generally used types of lighting equipment is the white glass enclosing globe. If a lamp is placed in one of these units it is estimated that on the average the globe will absorb something of the order of 15 to 25 per cent of the 81 per cent radiant energy reaching it from the lamp filament and nearly all of the 8 per cent reaching it from the bulb. Hence, when the lamp bulb is placed in a white glass enclosing globe there will be 60 to 65 per cent of the energy dissipated in radiant form. The remaining 35 to 40 per cent of the energy is dissipated from the lamp and enclosing globe by conduction and convection. Theoretically, it is only this heat that can be removed from the lighting units by a circulating air system and some attention has been given to this possibility in order to reduce the heating effect in a room.

If no attempt is made to intercept and carry away any of the heat generated by the lamp, the question arises as to what portion of it is actually effective in raising the room temperature or in augmenting the load on the cooling system in an artificially cooled building. At first thought it would appear that the heat equivalent of the entire energy input to the lamps, amounting to 3.415 Btu per watt per hour, must inevitably be regarded as being effective. In fact, that assumption is ordinarily made in engineering calculations for a cooling system. However, a close examination reveals that a fairly good safety factor is being applied.

It is to be remembered that the engineer is concerned primarily with heat as it affects human comfort, and this may be either in the form of radiant heat or as heat in the surrounding air. Radiant heat which comes from a lighting unit or is re-radiated from the walls or ceiling is instantaneously effective in producing a feeling of warmth. Its effect can be counteracted by maintaining a lower air temperature in a building which is artificially cooled. However, not all of the radiant heat from the lamps becomes a load upon the cooling system. A portion of it passes through the windows to the out-of-doors and some of that which strikes the walls, ceiling, and floor is absorbed by the building structure and partially dissipated during the cooler night hours.

The ceiling height of the room has an important bearing upon this problem. The direct radiant heat from the lights will be much more noticeable in a low-ceiled room because its effect varies inversely as the square of the distance of

the human occupants from the heat source. Another effect of the ceiling height is due to stratification of the air. The heat from the lamps which is intercepted by the glassware surrounding them, amounting to 35 to 40 per cent of the total energy of the lamps, causes the air to be heated surrounding the lamp and forms a stratum of very warm air near the ceiling. If this stratification is not disturbed by forced air currents it will not affect the comfort of the occupants below, especially if the room has a high ceiling. If, however, the cooling system is designed to introduce the air at high velocity several feet above the floor, the stratification is disturbed and the lighting fixtures are swept by the

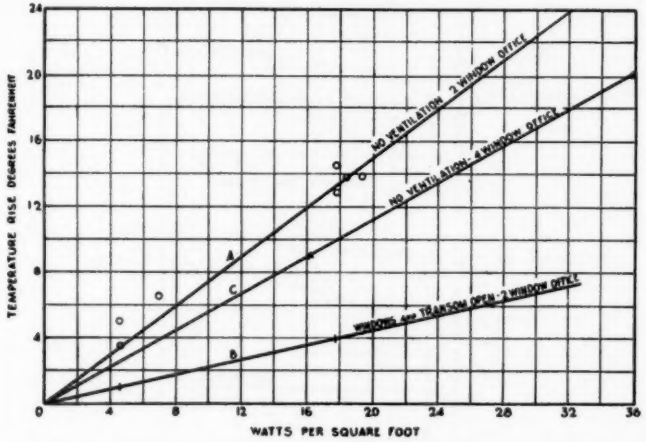

FIG. 4. RELATION BETWEEN THE TEMPERATURE RISE DURING A 7-HOUR PERIOD AND THE WATTAGE CAPACITY EMPLOYED FOR LIGHTING IN TWO TYPICAL OFFICE SPACES

currents of incoming cool air and their heat, absorbed by the incoming air, becomes a part of the load on the cooling system.

ROOM TEMPERATURE RISES

In order to investigate the relation between the temperature rise and the wattage capacity of the lighting system in a typical commercial office a space was selected in an office building where the lighting capacity could be varied. This particular space had approximately 300 sq ft (17 ft by 17½ ft) of floor area with an 11-ft 9-in. ceiling, and had two outside windows, each of which was approximately 3 ft wide and 6½ ft high. The results of the investigation are shown in Fig. 4, Curves A and B. Curve A represents the temperature rise for various wattage capacities of lighting when the room is closed as much as possible by keeping the windows and doors shut. Curve B represents the temperature rise on typical summer days when the office is ventilated by raising the lower sash of the windows and opening the transom over the door. For these tests the thermometers were suspended in a vertical position near the

center of the room with the bulbs 3 ft above the floor. Temperature readings were taken for a period of 7 hours when a condition of equilibrium was being approached; that is the heat dissipated from the room was nearly equal to the heat input and there was consequently but little further rise to be expected.

From the curves it will be observed that in the typical closed room with no artificial or natural ventilation the temperature rise during the 7-hour period due to the lighting system is approximately ¾ F for each watt per square foot of electrical energy consumed (Curve A). On the other hand, with the win-

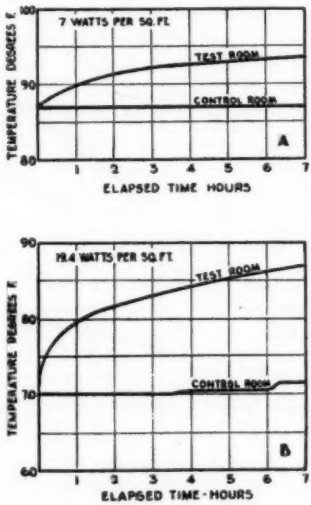

FIG. 5. CHARACTERISTIC CURVES INDICATING THAT APPROXIMATELY HALF THE TOTAL TEMPERATURE RISE FOR THE 7-HOUR PERIOD OCCURRED DURING THE FIRST HOUR

dows and transom open for natural ventilation on typical summer days the temperature rise is of the order of 0.22 F for each watt per square foot of lighting capacity (Curve B). The lighting equipment consisted of four open-top, semi-indirect units at the ceiling supplemented by indirect floor standards to provide the higher capacities.

Temperature readings in the test room were compared with those observed in an adjacent control room where no artificial lighting was used. The control room temperatures always remained practically constant during the test period. The rise in temperature in the test room was taken as the difference between its initial and final readings, and then this difference was corrected with any change which may have taken place in the control room.

The curves in Fig. 5 illustrate how the temperature in the test room rose

with respect to the elapsed time during the 7-hour period when 7 w per square foot were used (Curve A) and when 19.4 w per square foot were used (Curve B). From these typical curves it is noted that approximately half the total rise occurred during the first hour.

The curves in Fig. 6 indicate that the outdoor temperature had little effect on the test room temperature rise during the 7-hour period when the windows were closed. It should, however, be mentioned that on the colder test days both the control room and the test room received heat from adjacent heated areas in the building so that comfortable basic temperatures were maintained. The test room curves represent conditions where 18 to 19 w per square foot were

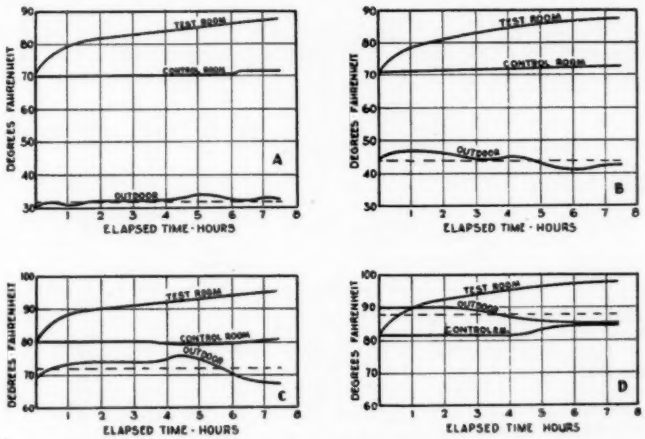

FIG. 6. CURVES A, B, C AND D SHOW TEMPERATURE RISES WHEN 18 TO 19 W PER SQUARE FOOT CAPACITY WAS USED FOR LIGHTING AND WITH OUTDOOR TEMPERATURES OF 32, 44.5, 72 AND 87.5 F RESPECTIVELY

employed (approximately 75 footcandles of illumination) with outdoor temperatures ranging from 32 to 87 F. Temperature rises of the order of 13 to 14 F were obtained.

On the other hand, with 18 to 19 w per square foot for lighting the room when the windows were open on a typical summer day the rise in temperature was of the order of 4 to 5 F for the 7-hour period. This is shown by Curve B, Fig. 4.

For general lighting throughout interiors such as stores, offices, and factories the electrical capacity in actual use at the present time is 1 to 2 w per square foot of area. With 2 w per square foot it is observed from Fig. 4 that the temperature rise in the closed room is 1½ F while in the ventilated room during the summer time it is only about ½ F. These temperature rises are so small that for uncooled buildings little notice has been given to the heat generated by the artificial lighting system.

A few typical examples of the relative importance of the several items

which constitute the cooling load in different kinds of buildings are shown in Table 2. These are values which were calculated in the process of designing the cooling systems. The first building in the table is a sales office of an electrical company. The show windows are not partitioned from the salesroom and the heat from the high level window lighting must therefore become a load on the cooling system. The building is shaded by adjacent buildings so that the sun never shines through the windows and the external load is therefore small. This combination of conditions causes the lighting to be 50 per cent of the total calculated load.

At the other extreme is the theater, in which the lighting is almost negligible

TABLE 2—TYPICAL EXAMPLES SHOWING THE RELATIVE IMPORTANCE OF FACTORS AFFECTING THE COOLING LOAD IN DIFFERENT CLASSES OF INTERIORS

TYPE OF INTERIOR	LIGHTING, WATTS PER SQ FT OF FLOOR AREA		RELATIVE FACTORS IN PER CENT				
	Installed	Assumed in Use	Lighting[a]	Solar Radiation Through Windows	Heat Conducted Through Walls and Windows	Infiltration of Outside Air	Occupants[b]
Sales office of electrical company with show window lighting	6.9	4.4	50	0	2	46	2
Office building							
With awnings	7.0	3.1	37	0	11	37	15
Without awnings	7.0	3.1	26	33	4	26	11
Office building with glass block walls	4.5	1.8	31.4	22.7[c]	4.6	28.2	13.1
Cafeteria	0.9	0.9	9	0	6	28	54
Theater	...	2.6	5	0	5	45	45
Research Laboratory	2.0	0.7	8.6	...	...	...	...

[a] Includes fan power load.
[b] Includes latent heat load.
[c] Includes heat gain by conduction through the glass blocks.

as a source of cooling load. The load imposed by the occupants and by the introduction of outside air constitutes most of the total load. In the case of one of the office buildings, the use of awnings eliminates the solar radiation through the windows as a source of load to the cooling system, thus increasing the relative effects of the lighting and other factors.

The values in Table 2 represent the maximum cooling load, which is used in selecting the size of the apparatus. It is of course true that, on the cooler days in summer, the load caused by the lighting, though no larger in absolute quantity, is a greater proportion of the total. The figures referring to the lamps are based on the assumption ordinarily made that the entire heat equivalent of the energy input to the lamps which are assumed to be actually in use becomes part of the cooling load. Therefore these values show the maximum

effect of the lighting on the cooling system. It may then be concluded that the lighting is responsible for a part of the cooling load varying between 5 and 50 per cent of the total, depending upon the type of building. Recognizing this fact, thought has been given to ways of eliminating the heating effect from lighting units in some manner such as having the heat diverted directly from the source into a ventilating duct where it will be dispensed with or utilized as may be desired.

DIVERTING HEAT AND ABSORBING RADIANT ENERGY

In order to take advantage of the natural tendency of heat to rise into higher altitudes, an experimental lighting system was installed which employed four 20-in. enclosing glass globes with hollow fitters connected to openings in a false ceiling. Fig. 7 illustrates diagrammatically the installation which was

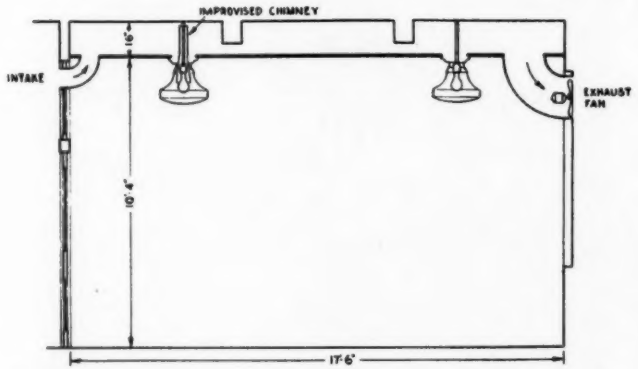

FIG. 7. CROSS-SECTIONAL VIEW OF TYPICAL OFFICE USED IN TEST SHOWING FALSE CEILING WITH OPENINGS ABOVE THE LIGHTING UNITS AND LOCATION OF INTAKE AND EXHAUST FOR VENTILATING AIR

made in a typical office room. An exhaust fan was used to remove the air from the space above the false ceiling; the volume of air removed and the temperature rise between the intake and the exhaust air were observed. The results of the test are shown in Table 3, with a total of 4163 w actual consumption at the sockets and with the room doors and windows closed.

Circulating Air Through a False Ceiling

The data from Tests 1 and 2 indicate that about 30 per cent of the heat generated by the lighting units can be removed by permitting the heated air to rise above a false ceiling where it can be carried away by a ventilating system. Although a greater portion of this heat was removed directly from the lighting unit, yet a small portion of it was no doubt conducted from the room through the false ceiling into the ventilating air. It was felt that further experimental work should be done to obtain more accurate and reliable

data on the maximum amount of heat which can be removed directly from a light source by circulating air through it, and therefore another series of tests were made, as will be described later.

By removing approximately 30 per cent of the heat the temperature in the room increased from 80 to 91.5 F, resulting in a rise of 11.5 F, during the 7-hour period. Under similar conditions but without the ventilating system operating above the ceiling, the temperature in the room increased from 69.5 F to 88.5 F, a net rise of 19 F during the 7-hour period. In this latter test it is believed that the 19 F rise is 4 to 5 F higher than it should have been. This opinion is based upon the fact that for this particular test the initial temperature was 69.5 F while the control room registered 75 F. The test should have been repeated with approximately the same starting tem-

TABLE 3—DATA AND RESULTS OBTAINED FROM TESTS CONDUCTED IN ROOM SHOWN IN FIG. 7

	Test No.					
	1	2	3	With Chimney Sleeves on Lamp		
				4	5	6
Change of Air, cfm	442	342	No change	442	342	No change
Intake Temp., F	80.3	79.4	...	78.5	82	...
Exhaust Temp., F	89.5	90.7	...	85.5	89.4	...
Ventilation Air Rise, F	9.2	11.3	...	7	7.4	...
Btu/min Removed	72	68.5	...	54.7	44.8	...
Btu/min Generated	237.3	237.3	237.3	237.3	237.3	237.3
Per Cent Btu Removed	30.3	28.9	...	23.1	18.9	...
Test Room Temp. Rise after 7 hours, F	11.4	11.7	19	10.5	11.7	15.6

perature for the two rooms but the equipment was dismantled before the discrepancy was noted. In a majority of the other tests the difference between the starting temperatures of the two rooms was less than 1 F and in no case did it exceed 2 deg.

In attempting to remove heat from the lighting units in the foregoing manner several tests were made with cylindrical and conical-shaped tubes (see Fig. 7) inside the enclosing globes over the lamps to create a greater flow of air upward around the lamp bulbs. The results of this investigation are given under Tests Nos. 4, 5 and 6 in Table 3 which indicate that the flow of heat from the lamp inside the enclosing globe was actually retarded rather than assisted by the improvised chimneys. For example, in comparing the results of Test No. 4 with No. 1 it will be noted that only 23.1 per cent of the generated heat was removed when the chimney was used, while without it 30.3 per cent was removed. The test room temperature rise was found to be about the same in both cases. The opening into the ceiling was 4⅞ in. in diameter and, when a thin-walled cylinder 3 in. in diameter with conical

shaped lower section was placed through it and around the lamp bulb, the resistance to the air flow was apparently too severe to be overbalanced by the theoretically greater draft action which should have been created by the chimney. The cross-sectional area of the chimney was made approximately one-half the total area of the opening so that the air flow would be as free as possible.

Considerable time and effort was spent to determine the most effective size as to diameter and length and the best shape of the chimneys as well as their proper location in the lighting units. On this phase of the investiga-

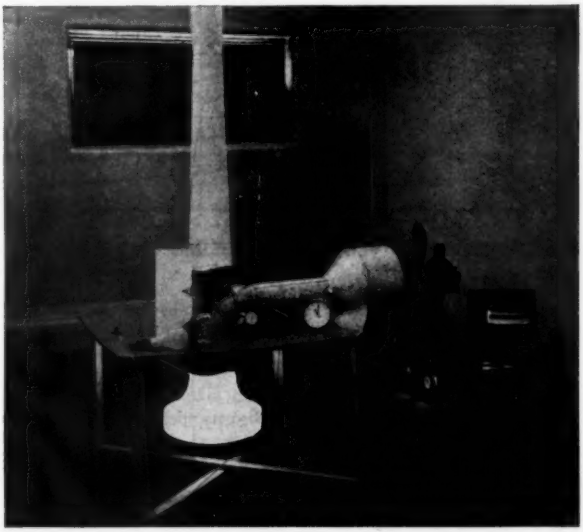

FIG. 8. ARRANGEMENT OF EQUIPMENT TO DETERMINE THE MAXIMUM AMOUNT OF HEAT REMOVABLE BY FORCING AIR THROUGH AN ENCLOSING GLOBE WHEN SUSPENDED IN A ROOM HAVING A CONSTANT TEMPERATURE

tion the effectiveness of the chimney was judged by observing the temperature at a point on the side of the enclosing globe. The point for observing the temperature was taken about half way down the side of the globe so that none of the various sizes and shapes of sleeves used intercepted the direct radiant energy between the light source and the thermometer. The bulb of the thermometer was placed in direct contact with the enclosing globe and held in the same position throughout the investigation. Then, for any given condition of ventilation in the room or above the ceiling, temperature readings were observed on two lighting units, one unit being the control unit and the other one the test unit with the sleeve in it. The readings indicated that different sizes and shapes of sleeves had very little effect on the tem-

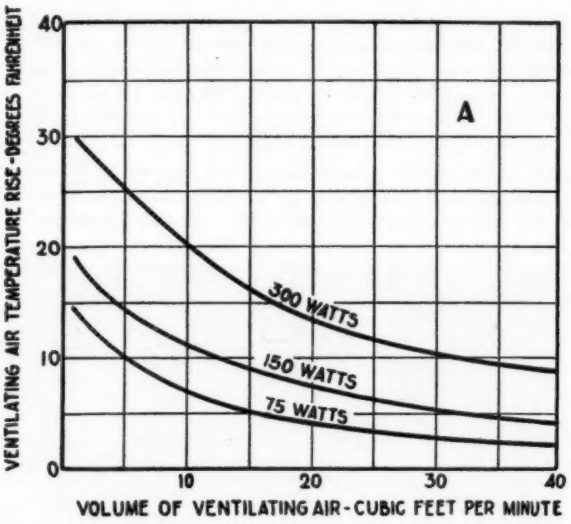

FIG. 9A. RELATION BETWEEN VOLUME OF VENTILATING AIR AND ITS TEMPERATURE RISE WHEN LAMPS OF THE 75, 150 AND 300-W SIZES ARE USED IN EQUIPMENT SHOWN IN FIG. 8

perature of the globe, assuming a minimum size of sleeve of 3-in. diameter and a sufficient length to extend from slightly below the socket to a point well above the false ceiling.

Tests Nos. 4 and 5 having forced ventilation above the ceiling show room temperatures about 5 F lower than Test No. 6 which represents a similar test but without forced ventilation.

MAXIMUM HEAT REMOVED BY FORCED VENTILATION

As mentioned in the foregoing, it was questionable as to just how much of the approximately 30 per cent heat removed by Tests Nos. 1 and 2 was due to ventilating the lighting units and how much was due to conduction through the ceiling. Further tests were therefore made with equipment as shown in Fig. 8 to determine the maximum per cent which can be removed by circulating air through lighting units of a type and design in general use suspended below a ceiling. This equipment consisted of a 14-in. white glass enclosing globe having a hollow fitter through which intake and exhaust air ducts were brought to the lamp. For the investigation the enclosing globe was suspended in a room where the temperature was held constant while different volumes of air were passed through the globe in which different sizes of lamps were burned. The curves in Fig. 9A show how the ventilating air temperature increased as the volume of the ventilating air decreased for lamps of the 75, 150 and 300-w sizes. From the temperature

rise and volume of the ventilating air the heat removed was computed. Fig. 9-B shows the heat removed in per cent of that generated by the lamp for different volumes of ventilating air. It will be noted that these three curves practically coincide when 35 to 40 cfm of forced ventilation are used. Under this condition it appears impossible to remove by a current of air more than about one-third the total heat generated by the lamps. This amount checks closely with the theoretical percentage mentioned earlier in this paper.

Another test was conducted with the equipment shown in Fig. 8 by placing the 14-in. enclosing globe in a cubical box 34 in. on a side as illustrated in Fig. 10. The cubical box was made of ½-in. thick insulating board which is a comparatively poor conductor of heat. The object of this test was to compare the air temperatures in the box when the lamp was ventilated, with those without ventilation. Lamps of the 40, 75, 150 and 300-w sizes were used with approximately 75 cfm of ventilating air passing through the enclosing globe. In Fig. 11 the solid-line curves represent the temperature rise during a 4-hour period when the lamp was ventilated and the dotted-line curves represent the temperature rise during a similar period with no ventilation. A 4-hour period of operation was chosen as it was found that it required about this length of time for the temperature in the box to approach a maximum. The data in Table 4 based on temperature readings at the end of the 4-hour period indicate that the box temperature was approximately 40 per cent lower when the lamp was ventilated than it was with no ventilation. If this box could be considered as a small room having ½-in. thick

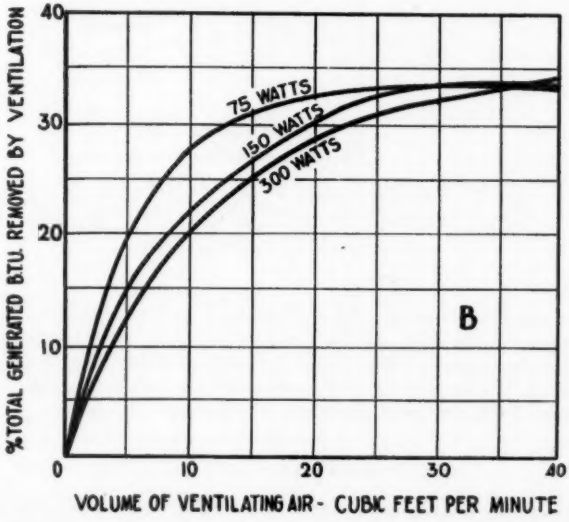

FIG. 9B. RELATION BETWEEN VOLUME OF VENTILATING AIR AND PER CENT BTU REMOVED AS COMPUTED FROM DATA IN FIG. 9A

insulating board as side walls, floor and ceiling, and no openings (and it is assumed that all the radiant energy must be absorbed somewhere within the room), then it is found that a maximum of approximately 40 per cent reduction in temperature can be obtained by circulating cooled air around the light source.

Further data from the box tests permitted the determination of the heat removed from the lighting unit by the air duct. When approximately 75 cfm were used in the air duct, the temperature rises of the air at the end of the 4-hour period are as given in Table 4. From these data the heat removed from the lamp was computed and found to represent approximately 60 per

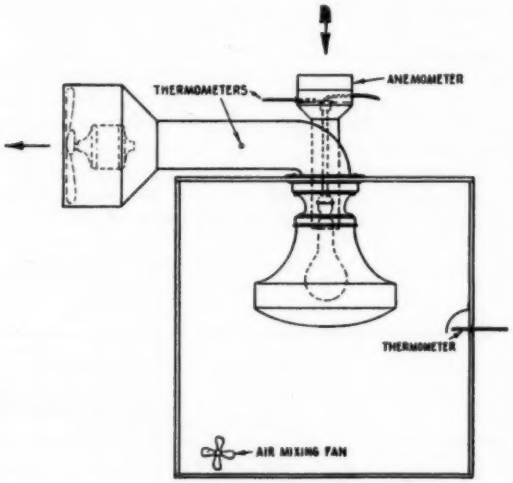

FIG. 10. CROSS-SECTIONAL VIEW OF INSULATING BOARD BOX AND EQUIPMENT EMPLOYED FOR COMPARING THE TEMPERATURE RISE IN THE BOX WHEN THE UNIT IS VENTILATED AND NON-VENTILATED

cent of the total generated. This percentage was considerably higher than that found in the previous test for the unit (Fig. 8) when operated in a constant temperature room. This difference was probably due to the abnormal conditions of having the lighting unit confined within the box where the temperature of the enclosing globe was obviously much higher.

HEAT RESISTING AND ABSORBING MEDIA

It is generally known that water will readily absorb practically all the long-wave infra-red radiation, that is, wave lengths longer than 1.4 to 1.6μ (microns). On the other hand, water transmits a high percentage of the energy appearing in wave lengths shorter than 1.0 to 1.2 μ which includes the

visible energy. Fig. 12 shows the relative energy at different wave lengths as generated by a 500-w gas-filled incandescent lamp. Superimposed on this curve is a transmission curve for energy passing through 1 cm (centimeter) of distilled water. From this it is obvious that lighting units with cold water jackets instead of cold air would greatly facilitate the elimination of heat immediately at the light source. A light source adopting this feature was developed several years ago [2] and recent designs based on the original features have found practical application for the lens-type photographic enlargers.

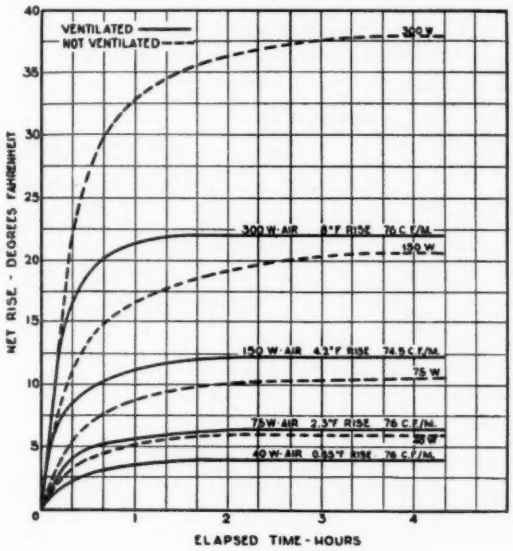

FIG. 11. CURVES SHOWING NET TEMPERATURE RISES DURING A 4-HOUR PERIOD IN TEST BOX WITH FOUR DIFFERENT SIZES OF LAMPS IN LIGHTING UNIT WHEN VENTILATED AND NOT VENTILATED

The unit consists of a distilled water jacket surrounding the lamp and a submerged coil through which cold water is continuously circulated. Tests on such a device employing a 1500-w projection type lamp indicate that 75 per cent of the radiant energy is absorbed by the water while the sacrifice in total light output is only of the order of 20 per cent; although this method of dispersing the heat from an artificial lighting system is entirely practical under certain conditions, yet its general application for large interiors appears somewhat remote at the present time.

Another medium for absorbing radiant energy from incandescent lamps has been developed [3] in a heat-resisting, heat-absorbing glass having a 2-

[2] Water Cooling of Incandescent Lamps, by N. F. Gordon (*Society of Motion Picture Engineers Transactions*, Vol. 14, No. 3, p. 332).
[3] Color Glass Filters (Corning Glass Works Bulletin No. C-112).

millimeter thickness which will absorb 79 per cent of the total energy generated from a 500-w projection lamp. The light transmission of this special glass is still maintained fairly high, being approximately 75 per cent. Plates of this glass are available for use in false ceilings above which artificial lighting systems can be installed. With such a system a circulation of air could be produced to cool the glass plates and thereby to provide a method of eliminating a high percentage of the radiant energy from a lighting system before that energy is set free in an interior.

TABLE 4—DATA AND RESULTS OBTAINED FROM TESTS WITH A 14-IN. WHITE GLASS ENCLOSING GLOBE IN A 34-IN. CUBICAL BOX

LAMP SIZE WATTS	NON-VENTILATED LIGHTING UNIT NET TEMP. RISE IN BOX, F	VENTILATED LIGHTING UNIT (APPROX. 75 CFM)			BTU PER HOUR GENERATED BY LAMP	PERCENTAGES	
		Net Temp. Rise in Box, F	Temp. Rise in Air Duct, F	Btu per Hour Removed by Air Duct		Temp. Rise in Box Ventilated to Non-Vent.	Heat Removed to Heat Generated
300	37.9	22	8	608	1024.5	58.0	59.4
150	20.7	12.3	4.2	326	512.3	59.4	63.5
75	10.5	6.3	2.3	182	256.1	60.0	71.0
40	6.2	4.0	0.85	67	136.6	64.5	49.3
Average						60.5	60.8

PRACTICAL APPLICATIONS

Many lighting systems have in the past been installed with ventilation ducts to eliminate the heat. Such systems have always been associated with high levels of illumination, 50 to 100 footcandles or more, as generally used in display windows, hospital operating rooms and photographic studios. An example of the removal of lamp heat by air circulation is illustrated in Fig. 13. This shows a sales office of an electrical company having high level window lighting in which the windows are not partitioned off from the sales space as is usually the case. The heat generated by the lamps would therefore be an expensive factor in the cost of operating the cooling system installation.

The lighting units are enclosed in metal boxes, each of which has a duct which supplies cool air drawn from the basement. The warm air is discharged to the out-of-doors during the summer when the artificial cooling system is in use. In winter, by a change of dampers, the heat is diverted to the second floor of the building where it is utilized for part of the heating requirements. Means are provided for installing a circulating fan but the gravity circulation alone appears to be sufficient. This installation was made partly for economic reasons in view of the cost of summer cooling, and partly to permit better control of temperatures during the heating season.

A recent article [4] gave details of a ventilated lighting system in a Philadel-

[4] Removing Heat from High Intensity Lighting (*Electrical World*, April 15, 1933, p. 480).

phia display room having dimensions of 40x110 where 40 footcandle illumination was supplied from eight large alabaster diffusing glass units each containing thirteen 500-w incandescent lamps and four 450-w mercury lamps, making a total of 8.3 kw. With a heat dissipation of 15.5 w per square foot of floor space, the discomfort resulting therefrom had to be prevented by the air-conditioning equipment, which would have entailed with this lighting load an additional investment and operating expense that was unwarranted. Consequently a partly closed cylinder of heat-absorbing glass was placed around

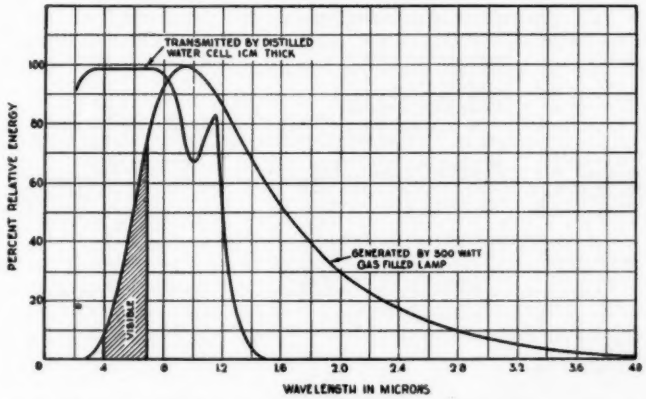

FIG. 12. RELATIVE ENERGY AT DIFFERENT WAVE LENGTHS FROM A 500-W GAS-FILLED INCANDESCENT LAMP AND THE PER CENT OF THIS ENERGY TRANSMITTED THROUGH 1 CM OF DISTILLED WATER

each incandescent lamp, through which air was drawn by a ventilating duct system connected to each fixture. It was found necessary to operate the ventilating fan for at least 5 min after the lights were turned off to avoid breakage of the alabaster diffusing glass by the sudden storage of heat, which would otherwise have occurred. The ventilator withdrew the air from each fixture at the rate of 600 cfm, discharging it outdoors in warm weather. When heat was needed in the building the air withdrawn from the fixtures was delivered to the air-conditioning system in such proportions as required. This arrangement reduced the cooling cost of the air-conditioning system one-half and the steam-heating system was only used on the very coldest days.

Another practical solution of eliminating heat was described [5] in connection with a clothing store located in Atlanta, Ga., which was already air cooled. The store was relighted by increasing the lighting load on the first floor from 4000 to 22,000 w. Due to the fact that the first floor of the store was air conditioned, the problem of increasing the lighting load from 4 kw to 22 kw

[5] Atlanta Store Acknowledges Added Business Through Improved Lighting (*Magazine of Light*, May, 1937, p. 20).

without affecting the capacity of the air conditioning equipment was overcome by building the fixtures (two luminous projecting beam type) the entire length of the area to be lighted and by placing a filter on one end of the fixture and a positive pressure fan on the other end. The amount of additional heat was eliminated by this design. The fan was separately controlled so that in the mild winter months it could be cut off, and only on a few occasions has the heat from the lighting fixture been insufficient to heat the store. Care should be taken in the construction of the fixture to make it fairly tight so

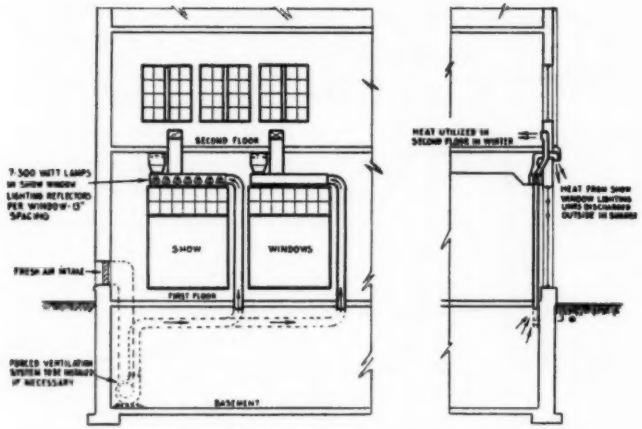

FIG. 13. SKETCH SHOWING VENTILATING SYSTEM FOR SHOW-WINDOW LIGHTING; DISPENSING WITH HEAT IN SUMMER AND UTILIZING IT IN WINTER

that it will act as a duct and extreme care must be taken in selecting the size of the ventilating fan.

In any room where warm air is displaced by cool air, it would seem logical to those who are inexperienced in the technique of air conditioning that the cool air should enter the room near the floor and the warm air be withdrawn at the ceiling. Quite the reverse is the modern cooling system which is designed to have the incoming air enter the room near the ceiling. This, of course, has certain advantages, the most important of which is the elimination of drafts and uncomfortably cool conditions for the occupants of the room.

The method of reheating the cold air to a comfortable temperature before it is released is a subject of much controversy. In general, the reheating is accomplished by mixing the dehumidified cold air with warm air which has either been by-passed around the cooling coils or which has been obtained from outdoors. In either case the volume of warm air must be sufficient to properly temper the air before it reaches the occupants of the room. In many cases at least a part of this tempering is accomplished by releasing

the dehumidified cold air near the ceiling and directing it around the lighting units where an adequate amount of heat is absorbed. Estimations indicate that all the sensible heat generated by a lighting system could be used for this purpose unless it exceeds about one-third the total sensible heat gain for the interior.

NEW LIGHT SOURCES

The curves in Fig. 1 show the gradual increases in efficiency for incandescent lamps which have been made during the past two decades. Although it is expected that further advances will be made, yet it is recognized that there is a definite limit to them because the efficiency of a lamp depends on the operating temperature of the tungsten filament, which in turn is limited in order to keep a safe distance below its melting point.

A new outlook with comparatively unlimited possibilities has recently appeared in the form of electric discharge lamps employing mercury vapor. These Type H lamps have entirely different operating characteristics and color quality of light from incandescent lamps, but they are suitable for a wide range of industrial and commercial lighting applications. They are now available in only two sizes, the 400-w (Type H-1) and the 250-w (Type H-2). Of outstanding importance is the fact that the 400-w lamp has an efficiency of 40 lumens per watt, and the 250-w lamp has an efficiency of 30 lumens per watt. These light output efficiencies can be compared with the 19 to 20 lumens per watt obtained from incandescent sources of an equal wattage. Mercury lamps (Type H) will therefore provide illumination with a heating effect which would at first appear to be considerably less than that required from an incandescent system. However, their color when used alone is not generally acceptable for stores and offices. Hence, for these interiors, mercury lamps are usually combined with incandescent lamps so that the combination gives substantially a daylight effect. Where this is done there is usually no saving in wattage.

The color of the light emitted by mercury lamps has the characteristic mercury vapor line spectrum in contrast to the continuous spectrum of the incandescent lamp. This means that all the light from the mercury arc is represented by only a few bands which produce yellow, green, and blue light. Red is practically absent. The result is that although the light itself has a bluish-white cast, materials of certain colors such as the reds are not recognized, while yellows and greens are emphasized. However, in the work spaces where whites, greys and other neutral tones predominate, Type H mercury lamps are satisfactory.

The operating characteristics of mercury lamps are such that each lamp requires a special transformer or regulating device for starting and operating. Also these lamps require several minutes to attain full brilliance after being turned on.

Another type of lamp now under development which promises to find application in auditoriums, theaters, restaurants, etc., where lumiline incandescent lamps are now being used, is the fluorescent lamp, which is in effect a mercury lamp similar to the Cooper-Hewitt but smaller. The lamp itself it not very efficient, but inside the tubular bulb there is a great deal of

shortwave, ultra-violet energy which does not appear in the form of light because it is generated below the visible spectrum. Not long ago it was discovered that when a powder which fluoresces under ultraviolet radiation is placed inside a mercury lamp, the powder turns the short wave energy into visible energy which produces a considerable amount of light. Since this discovery, there has been a search for powders that would produce white light efficiently and also other colors, for decorative purposes. Present developments indicate extraordinary increases in efficiency over that obtained

TABLE 5—APPROXIMATE RADIANT ENERGY FROM INCANDESCENT AND MERCURY TYPE H LAMPS[a]

LIGHT SOURCE	LIGHT OUTPUT (LUMENS)	RADIANT ENERGY (PER CENT)	WATTS DISSIPATED IN RADIANT ENERGY	RADIANT ENERGY (WATTS) FOR 10,000 LUMEN LIGHT OUTPUT
500-w incandescent lamp.....	10,000	90	450	450
400-w mercury.....	16,000	65	260	163
250-w mercury.....	7,500	70	175	233
15-w fluorescent (white experimental).....	450[b]	50	7.5	166

[a] Evaluated in watts dissipated for equal light production of 10,000 lumens.
[b] Assumed output.

from the present lumiline incandescent types. Fluorescent lamps are not commercially available as yet.

RADIANT ENERGY FROM MERCURY LAMPS

In analyzing the distribution of energy dissipated from 300- to 500-w incandescent lamps, it was found that approximately 90 per cent of it occurred in the form of radiant energy. In comparison to this about 70 per cent is found for the 250-w Type H lamp and 65 per cent for the 400-w Type H. The fluorescent lamp (15-w 18-in. white experimental) dissipates about 50 per cent of its heat equivalent in the form of radiant energy. Using these percentages the heat dissipated in radiant form can readily be computed for each light source (see Table 5). The last column in Table 5 shows the comparative radiant energy in watts when adjusted to a 10,000 lumen light output basis.

The radiant energy would, of course, be reduced somewhat if the lamps were placed in lighting units which would intercept and re-radiate it. Although no definite data are available, it is estimated, for example, that when a 400-w mercury lamp is placed in a lighting unit less than half its total energy will be dissipated in radiant form. In other words, a comparatively high percentage of the energy would be conducted and convected away from a mer-

cury lamp in a lighting unit. Mercury lamps of the future may not only have an advantage over incandescent lamps from an efficiency point of view, but also from the cooling and ventilating viewpoint, because a greater proportion of their heat energy may be carried out of the building by air drafts around them when placed, for example, above skylights or behind built-in glass plates. In some cases, however, air drafts may affect the operating efficiency of mercury lamps, but experience data along this line are lacking.

CONCLUSIONS

For lighting systems employing the usual type of luminaries placed below the ceiling, it is mechanically possible to install a ventilating system with air ducts to the units so that a maximum of about one-third the total generated energy can be removed. With the so-called *built-in* types of lighting equipment where a free circulation of air is provided around the lamp, the amount of heat carried off will depend upon the heat transmission characteristic of the glass through which the light enters the room. If this glass is of the heat-resisting heat-absorbing type and is itself cooled, then at least twice as much can be removed by the ventilating air as found possible from the usual types of lighting units suspended below the ceiling.

If none of the energy generated by the lighting is carried off by a ventilating system it is estimated that from 50 to 90 per cent of it will eventually become a part of the cooling load. Inasmuch as a maximum of only one-third the total energy can be removed from the usual types of lighting equipment, it is the opinion of the author that the cost of an elaborate auxiliary ventilating system for this purpose would not in general be economically justified for the present standards of lighting whose requirements, as indicated in Table 1, seldom exceed 5 w per square foot of floor area. In other words, for these general interiors it would be less costly to provide sufficient cooling capacity to offset the heat generated by the lamps as well as that from other sources.

However, for interiors where high levels of illumination are used which require 10, 15 or more watts per square foot of electrical capacity, careful consideration should be given to a separate ventilating system which not only carries off the readily removable one-third of the heat energy, but also makes use of heat-resisting glass to intercept a considerable portion of the radiant energy.

In the foregoing discussion the problem of removing heat generated by the lighting system is of importance in northern latitudes for only 3 to 4 months each year. During the other 8 or 9 months this heat may be of considerable value in contributing to the heating system for the building.

ACKNOWLEDGMENT

The author wishes to acknowledge his appreciation of the valuable suggestions received from several heating and ventilating consultants and particularly his indebtedness to J. H. Walker, of the Detroit Edison Co., for data and assistance and to the *Illuminating Engineering Society* for data obtained from their *Transactions*.

DISCUSSION

A. D. CAMERON[6] (WRITTEN): It has been no secret that an artificial lighting system presents a factor to be reckoned with by the air conditioning engineer. In some few cases its effect may have been under-estimated, more often it may have been over-estimated. The time for estimating, however, has long passed. Air conditioning is an exact science if all the contributing factors are known. This paper is an introduction to a joint study by lighting and air conditioning engineers of light and its by-product heat as it becomes a load on the air conditioning system, in order that the ultimate user may economically enjoy the benefits of health, comfort and personal efficiency which both systems provide. Mutual understanding in such a problem is always a long step toward the solution.

The paper states that in a lighting system requiring up to 5 w (watts) per square foot, the expense of ventilating the lighting fixtures does not seem justifiable. There remains, however, toward the upper end of this range a very appreciable load on the air conditioning system.

For higher levels of illumination, the lighting load is reported as being sufficient to justify the expense of ventilation.

This must not be taken as an assumption that the ultimate solution to the problem is to be passed along to the air conditioning engineer. The author acknowledges the joint responsibility of the lighting engineer and his opportunity for dealing with the problem at its source, the lighting unit itself. Admitting that the heat is proportionate to the wattage rather than to the amount of illumination, he has pointed out:

1. That increasing efficiencies of light sources provide increased illumination without increased heat.
2. That the type of lighting fixture used will also have an appreciable effect on the amount of illumination per watt. A high utilization of light can be obtained by selecting the type of fixture best suited to the task.
3. That the position occupied by fixture, that is, whether suspended or built-in, can be studied with profit since the latter system is capable of being conveniently ventilated.

The problem is then equally divided. The lighting engineer will develop his light source and his lighting fixture to obtain a high utilization of light with its corresponding minimum wattage requirement. The air conditioning engineer will study the most desirable location for the fixtures and decide when and to what extent ventilation is justified.

The ultimate user will choose his lighting system and his air conditioning system with reference to their related functions.

I am impressed with the careful research which produced the data for this paper. Further studies may even result in a new measurement factor which will express the performance of a luminaire in terms of both its lighting and its heating efficiency.

H. M. SHARP[7] (WRITTEN): This paper is a timely presentation in that it is eloquent of what is unknown, as well as bringing together in one publication a summation of what is known. Lighting and air conditioning are probably the most important contributions that science and engineering have made to the creation of healthful and natural conditions within the artificial environments that human beings are confined to. There is a community of interest between the two that can be greatly advanced by a continuing study of the intricacies of this problem.

I am very strongly of the opinion that the psychology of color, both in surroundings and light flux, must be extensively considered. The strong yellow sunlight streaming

[6] Vice-Pres., Holophane Co., Inc., New York, N. Y.
[7] Chairman, Committee on Lighting Service, *Illuminating Engineering Society*, Buffalo, N. Y.

through windows carries with it a liberal component of heat. The high levels of artificial illumination for today bring with them higher brightnesses of surroundings and work surfaces. Instantly, the association with nature is established. I have personal knowledge of where employment of cooler colors and the use of totally indirect lighting have eliminated sensations of heat, even though the Btu from lighting were increased. Again, investigations over a period of time, using recording thermometers, and under lighting systems screening all direct radiant heat from occupants, have disclosed no significant changes in sensible heat while the occupants insisted that lower levels of illumination were much cooler. These instances refer only to interiors where all conditions of lighting were in accord with the most advanced practice.

As engineers, we must first have facts, and then in applying them to the comfort of human beings, the intangibles should also have their day in court.

W. R. HEATH (WRITTEN): A matter that has been the subject of some indecision to the air conditioning engineer is covered in this paper. The virtue of ventilating lights has been recognized as a possible method of reducing the load where the power from lights is high, but no definite information as to the quantities involved has been available. The value of one-third the total generated energy is the same now as given by the same author in 1933. Whether or not the total wattage of unventilated light should be considered ultimately reflected in the form of room heat has not been proved. It seems logical to presume that it is not all so reflected; or only after a considerable time lag. Unfortunately, the measurement of room temperature rise gives only a non-quantitative comparison of various lighting systems which can not be converted into heat units as the rise depends on the size and construction of the room, the outside temperature and wind. Had the test room been held at the same constant temperature as the control room by some simple cooling system, whose heat absorbing capacity could be continuously measured, the heating effect of the lights could have been determined fairly accurately.

The suggested use of lights for re-heating might be convenient in certain cases, but conserves no more heat than steam re-heating; the light load is still present in the total heat load.

The conclusion that an elaborate ventilating system would only be economically justified when levels of illumination are high seems reasonable. It may be further qualified to cover only those rooms which also have a high total watt input. For example, 1 kw represents a heat output of 3400 Btu per hour. Thirty per cent of this would be about 1000 Btu per hour, so if the total light load amounted to 12 kw, there would be a possible saving by ventilating of 12,000 Btu per hour or one ton of refrigeration. This would represent a saving in first cost of the air conditioning system of, say $250, and a saving in power of approximately 1 kw. From the author's experiments, 12,000 Btu per hour saving would require about a 1000 cfm ventilating system. The cost of fan and motor together with duct work, control, and installation, would probably offset the $250 saving in first cost and not much further investment in special lighting fixtures or special building construction would be justified by the 1 kw power saving.

If, however, the total load were several times that mentioned, the saving both in first cost and power would be increased in proportion, but the cost of the ventilation system would increase at a much lower rate and the possible saving would be increased.

At present it would appear that the practical application of a light ventilating system in an air conditioned room would be confined to such applications as show windows and operating rooms with a possible field in large areas with unusually high light intensities.

C. M. ASHLEY (WRITTEN): Mr. Sturrock's paper represents a distinct contribution to the application end of air conditioning and should do much to clarify and reduce to quantitative terms the effect of artificial lighting on the air conditioning load.

The importance of this subject is increasing due to the modern trend to higher lighting intensities in merchandising and other applications. Frequently in designing new stores or renovating old ones the problem includes both a stepping up of the light intensities and air conditioning, and it is highly desirable to avoid a conflict between these two modernization mediums.

I would like to emphasize one phase of the problem on which the author did not dwell. This is the effect of radiant energy from artificial lighting on the feeling of comfort at a given temperature and humidity level. The motion picture studio presents this problem in its most extreme degree. In this application the concentration of radiant energy on the actors is extremely high, and it is necessary to maintain a very low air temperature in order to avoid extreme discomfort. However, the problem still exists to a lesser extent even with more moderate lighting intensities. For instance, the effect of a lighting intensity of 10 w per square foot may be to increase the temperature of a person's clothing subjected to this radiation as much as 5 to 10 deg. The effect on comfort is undoubtedly not as great due to evaporation and the circulation of air through the clothing, but it must nevertheless be quite appreciable. I believe this factor should be the subject of further attention and study.

I notice in the experimental observation, no provision was made to protect the thermometers from this direct radiant effect. Undoubtedly, the radiant effect is responsible for part of the temperature rise. However, since most of these observations are of comparative rather than absolute value, this would not seem to detract from the value of the results.

Another rather interesting phase of this problem has recently come within my personal observation. This is, the effect of artificial lighting on the temperatures in different parts of a single air conditioned zone. In an air conditioned space consisting of offices and drafting room it was found that, with a system balanced to give uniform temperatures without artificial light, the use of the drafting room lights on dull days caused a change of temperature between the drafting room and the rest of the conditioned space of as much as 5 deg, with probably an even greater difference in the feeling of comfort. An analysis of the lighting intensity in this case showed it to be somewhat above the optimum, so that when the lights were adjusted to a proper intensity the difference in observed temperature was approximately 3 deg instead of 5 deg.

W. F. FRIEND[8] (WRITTEN): Lighting as a component of cooling load for air conditioning is of particular concern to electric utility companies. Improvement in lighting systems and adoption of more adequate illumination intensities are actively being encouraged by them, for existing buildings as well as for new ones under design. In some cases, modern lighting standards can not be adopted unless accompanied by air conditioning to dissipate resulting heat load; conversely, when air conditioning is installed, it becomes practicable to bring the lighting up to present standards. Furthermore, utility companies for their own salesrooms and office buildings are adopting air conditioning as rapidly as the situation warrants; removal of heat due to high lighting intensities used by them is a major consideration in most utility-building installations.

Table 1, p. 218, presented by the author gives recommended footcandle intensities of illumination, designating these as minimum standards. The question arises, in deciding the cooling load imposed on a projected air conditioning system, how much allowance should be made for advances anticipated over the next few years, in the art of illumination—so that the installation may not prematurely become inadequate. The curve in Fig. 1, representing increase in lumens per watt from incandescent lamps, accomplished since 1916, supplemented with remarks on outlook for new forms of light sources, is informative, but it would be helpful also if data could be given to show the trend of per square foot wattages in representative buildings, during the past dec-

[8] Mech. Engr., Ebasco Services Inc., New York, N. Y.

ade or so; this might well be accompanied by a curve showing history of hourly lighting cost per square foot floor area or per million lumens, taking into account both electricity rates and lamp replacement costs.

Again referring to Table 1, electric demands (watts per square foot) shown in second column are between ⅓ and ½ the footcandles in first column. While general comments are given in the paper, relating to factors controlling the ratio, more specific statement would be valuable to air conditioning engineers. Presumably the type of lighting fixture employed, nature of interior decorative scheme and room furnishings, and use of floor or table lamps to supplement general illumination, are major factors. In addition, consideration should be given to character of maintenance work likely to prevail in building; with lamp replacements made only after burn-out occurs, or when redecoration of ceilings is postponed until pronounced dilapidation becomes apparent, lamp wattages must be substantially higher than if measured drop in illumination level is criterion for routine replacement and maintenance work.

After installed lighting wattage has been decided upon, it is necessary to determine how much of this will be in use at the time of maximum heat gain from other sources. Adequacy of natural lighting at the sides of building not exposed to the sun will usually determine this, and the situation in each case can be ascertained by survey of the premises. However, along sunny walls it is frequently observed that Venetian blinds are closed and the lighting system is operated at full capacity. Test data, comparing the heating effect from a lighting system with solar heat admitted through Venetian blinds or awnings adjusted in such a way as to eliminate need for electric lighting, would be useful for guiding the practices followed by room occupants.

To district steam and gas utilities supplying winter heat for office buildings, information in Figs. 4 to 6 is of considerable value, as indicating the extent to which use of their services may be affected by high lighting intensities. Roughly, 6000 Btu per cubic foot of net rentable building volume are required annually for heating office buildings in a climate having a 5500 degree-day heating load. If lighting of 5 w per square foot is in use during half the hours of occupancy throughout the heating season, about 1000 Btu per cubic foot are thus supplied, supplanting perhaps 15 per cent of steam otherwise needed.

In existing buildings having inadequate heating systems, or when admission of steam to radiators after night shut-off has been unduly delayed in extreme weather, it is not uncommon for room occupants to turn on high-intensity lighting systems, in order to benefit from the temperature-boosting effect obtained. This practice is equivalent to use of electric strip heaters, that have occasionally been inserted in distribution ducts of air conditioning systems, to attain zoned temperature control where inadequate provision for zoning was made in the original design. The author's data seem to indicate that for such purpose electric utilization by a lighting system has low effectiveness and, when lamp replacement costs are considered, usually will prove uneconomic.

The several methods described for direct removal of heat released by lighting fixtures involve a substantial outlay, and justification for this must be proved. Some of the methods proposed require increased story heights, as well as unusual construction of floor panels and suspended ceilings. Economic analysis must take these elements into account, comparing the penalty in increased fixed charges with the saving made in annual expense for the air conditioning system. Where lighting intensities exceed 8 to 10 w per square foot, as in show windows and other advertising displays, enclosures or hoods with separate ventilation will prove economical and in fact are now quite commonly used.

D. W. MCLENEGAN (WRITTEN): Mr. Sturrock's paper should be a very valuable one to air conditioning engineers, since it supplies authoritative data on several points which have previously been rather uncertain. The trend toward high lighting intensities is of concern to air conditioning engineers, not only as regards the increased capacity of air conditioning systems which must be planned to take care of the addi-

tional heat, but also in the control and zoning of such systems. Where the higher lighting intensities are used it is necessary to take into account the extent to which the lighting may be turned on in the various parts of an air conditioned building, in order to determine whether conditions in these various rooms or zones may be controlled from a single point or whether separate controls may be necessary, since the artificial lighting sometimes represents a significant part of the total cooling load.

One point which is difficult for the air conditioning engineer to judge is the percentage of the total installed lighting wattage which should be taken into account in the design of a cooling system at times when the other elements of the heat gain are at a maximum. This is particularly difficult, because the use of lights in any room is dependent on the tastes of the occupants. In the design of a new system it is usually important that a definite understanding be reached regarding the amount of lighting, if any, which will be in actual use on bright sunny days in the various parts of the air conditioned space.

It would be interesting to have Mr. Sturrock's comments on this point, if any specific comments are possible.

C. S. WOODSIDE:* I am particularly pleased to hear this paper presented before a gathering of this sort. I happen to be connected with the lighting industry and we are very anxious to have the heating and ventilating men cooperate in this problem. I trust that they are anxious to have us cooperate because if we go on our separate paths, doing our separate design work, there is bound to be trouble.

Several things Mr. Sturrock mentioned are of considerable importance in view of future work in this field. There is quite a trend toward down-lighting in stores, places where they need a high intensity of light. Previously we have recommended indirect lighting. That is a pretty expensive method of producing a high intensity, so with the advent of some of the newer designs in down-lighting, we can obtain high intensities with considerably less wattage, but when we do that we get a pretty intense beam not only of light but of heat. We may feel all right in the winter, but in the summer it is particularly objectionable.

The glass people have developed a number of types of heat-absorbing glass which work out very nicely in connection with this down-lighting equipment. So far all of this glass has a certain color and it affects the resultant color of the light, but progress has been fairly rapid, and some satisfactory glass is available in a number of different degrees of absorption. Naturally, if you absorb some of the heat you absorb some of the light, but fortunately we take out more heat than we do light, so there is a net gain.

I think that is the field which is going to show the most promise in the way of reducing the heat. It takes the heat out of the beam. It is still there around the fixture, but it is a sensible heat and can be withdrawn by the fan lighting system.

G. W. MARTIN: For the last seven years I have had the privilege of acting as a supervising engineer, with properties in New York and elsewhere, office buildings and hotels, and I have also had the privilege for 10 years of acting on the Research Committee of the Management Division of the Real Estate Board of New York.

The Better Light Better Sight Campaign was initiated, to the best of my knowledge, some years ago. I was made conscious of it by the fact that the electric load on our generating plants was increasing due to the change from the direct type of lighting to the indirect. The wattage consumption per square foot of rentable area increased, placing a load not only on the wiring system, most of which consisted at least in the tenant space of number 14 wire with a limiting capacity of 15 amp (amperes), although the Department of Water Supply, Gas and Electricity usually limits us to a 10-amp fuse even on new wiring.

As previously mentioned, the problem was first brought to my attention due to the increase in the electric load. On investigation, we ran into the Better Light Better

* Mgr., Commercial Engrg. Laboratory, Westinghouse Lamp Div., Bloomfield, N. J.

Sight Campaign, and we invited the chief engineer of the lamp department of the General Electric Co. and his assistant to one of our meetings. We discussed the matter with them, told them our problem and cited a table which I think was given in a pamphlet issued by the Company.

The one point to which we took the earliest exception was a recommendation of 10 footcandles on the floor of an elevator cab. We asked the chief engineer why that recommendation was made, and he said, "So a delivery boy can read the address on the package in the car and also to prevent any passenger from stumbling on the sill." The footcandles on the floors of elevator cabs in a number of modern buildings in New York run 2.5 or 3. None of the members of the Research Committee, Management Division, Real Estate Board of New York, numbering about 15, all of whom are engineers engaged in real estate operation, reported any accidents due to the low footcandle readings on the floor of their cabs.

Reference has been made to a low limit of 5 w per square foot for lighting. The chief engineer made the statement that he was recommending to architects a minimum wiring capacity of 5 w per square foot for lighting only in new buildings. We took issue with him at once. It not only involved a problem of existing buildings but it also involved, as it did in my case, an increase in load on the existing plant. We found that one of our tenants, who is nationally known as a paint manufacturer, with a complete indirect lighting system in their accounting department, and with an occupancy of one person to every 85 sq ft, which I would say was about average, was getting along very nicely with a wattage consumption of 2.6 w per square foot.

We had another talk by a member of the staff of a consulting engineering firm, who took the stand that a wiring capacity of 2 or 2.5 w per square foot would be sufficient for the proper lighting requirements.

The Research Committee of the Real Estate Board of New York has taken a middle course and we believe that the capacity of wiring for lighting should not be more than 3 w per square foot. The air conditioning load on the basis of square foot of rentable area is estimated at about 2 w.

The so-called base outlet load, to which machines are attached, would amount to something like 1 w per square foot. Following the recommendation of the lighting industry would give us 5 w for lighting, 2 w for air conditioning and 1 w for a base outlet load, making a total of 8 w per square foot. Some of the modern buildings erected in New York within the last six or seven years have a wiring capacity of only 3 w per square foot.

I just simply want to emphasize in conclusion that I do not believe it is necessary to have a wiring capacity of 5 w per square foot, that a capacity of 3 w per square foot would be ample.

I may add that the chief engineer and his assistant have both told me that they have found in their experiments that, if for a certain purpose 10 footcandles were not sufficient, the footcandles should be increased to 20, and if 20 were not sufficient the footcandles should be increased to 50 before you would appreciate an increase in lighting intensity. We have found under our experiments that the increases can be made to 12 and 15 without jumping to 20 and give perfectly satisfactory results.

G. E. MAY: I should like to mention something of a personal experience in New Orleans that I think and hope will be of interest to all of you.

We had the problem of relighting the sales floor of a large building. There was to be an increase of approximately 15 w per square foot without any appreciable addition of heat to this area. Mr. Sturrock's fine paper gave us a key to the solution of our problem.

As in the case of the Detroit building that Mr. Sturrock mentioned, our building had a blind area above a quadrilateral vaulted ceiling that served well as a plenum for exhausting heated air. In the center of each bay was mounted a direct type luminaire having concentric louvers for glare control, and a space between the outer shell and reflector for exhausting the heat.

The quantity of air to be exhausted through these luminaires was calculated equal to the outside air requirements for the entire relighted area when, as, and if it becomes air conditioned. Tests made in advance indicated that a 55 per cent total heat arrestance was possible with this quantity of air. This 55 per cent is not at all inconsistent with the figures mentioned in Mr. Sturrock's paper. Our arrestance was higher because the absorption of radiant energy in our fixtures was higher than in the type tested by Mr. Sturrock. Concentric louvers are not efficient, but worth the sacrifice in some cases in order to obtain control over glare.

We all believe that this installation has worked out very successfully, and I tell you of it to show you just how much may be accomplished by ventilation.

No. 1084

COMPARATIVE ANALYSIS OF OFFICE BUILDING AIR CONDITIONING SYSTEMS

By J. R. HERTZLER *(*MEMBER*), YORK, PA.

ACTUAL and potential field market studies for large commercial air conditioning equipment show office buildings to be of first importance. Each building warrants a careful analysis and study, if the proper system is to be installed to meet its requirements. A group of large, complete office building air conditioning systems have been analyzed in this paper for comparative purposes.

In order to show the wide variation in the types of equipment and methods of installation utilized, a group of 10 buildings, as indicated in Table 1, involving air conditioned areas of more than 1,000,000 sq ft have been used for analysis.

FACTORS AFFECTING DESIGN

In the designing engineer's approach to office building air conditioning, several features are of primary importance to insure satisfactory acceptance, some of which are outlined herewith.

Type of Building

In order to properly calculate the heat gain it is important to know the shape of the building, whether the building is of windowless design or insulated, and whether of tower construction with exposures on all sides or partially shaded. It is also necessary to determine, within the enclosure, the partition arrangements, whether temporary or permanent, whether the spaces are rented or unrented and whether the hours of occupancy are uniform or variable throughout the building.

Zoning

This includes the separation of those spaces with similar exposures or similar heat gains so that uniform atmospheric conditions can be produced for all variations of load.

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Apparatus Location

This factor is important from the standpoint of available space which can be sacrificed for the location of the air conditioning apparatus. This is closely allied with zoning and will have an important influence upon first cost as well as service and maintenance charges.

Air Distribution

The distribution of air is obtained in many different ways depending upon the architectural treatment and limitations. In practically all cases, proper air distribution will involve interior redecoration of existing buildings, thus increasing its importance to the owner.

Cooling Towers

Means of finally rejecting the heat removed from the conditioned spaces to the outside air is of extreme importance. Office buildings are located in metropolitan areas and consequently no cooling tower installation should render an inconvenience or nuisance to adjoining property. In selecting the location and type of cooling tower, the problems of (a) noise; (b) drift; (c) fogging; (d) appearance; (e) weight and structural supports; and (f) cost of operation are the principal considerations of the designing engineer.

Refrigeration

The medium furnished for dehumidification and cooling purposes must be of a type not to create hazards and should provide operating flexibility to economically and efficiently meet all conditions of load.

Heating

Combined with the cooling equipment should be means to provide, under automatic control, uniform and satisfactory conditions of temperature and humidity during the winter operation of the apparatus.

Features of Design

Always taken into account is the scope of automatic operation and control, types of air cleaning devices, possibility of need for independent humidity control in locations of excessive humidity or where moisture loads occur independent of dry or sensible heat gains, and other auxiliary items of peculiar importance in special cases.

BUILDING A—HOUSTON, TEX.

Type of Building. Existing office building is occupied by single tenant. Individual offices on each floor open to central corridors.

Zoning. Building is zoned east and west with individual floor control.

Apparatus Location. The 450 hp central water cooling plant is located in the basement. Central air washer for spaces up to and including eighth floor is located in the basement supplying dehumidified air to two local recircu-

lating fans on each floor, one for east zone and one for west zone. Separate dehumidifier for ninth floor is installed on roof to supply the director's room and executive offices having roof exposure.

Air Distribution on lower floors utilizes corridor ceilings for location of duct work for supply and return air. In office space, ducts are arranged on face of columns to appear as false beams—with columns off center. Overhead plaque system of distribution is used on ninth floor with ducts in hanging ceiling. Director's room air supply is directed upward through radiator enclosures.

Cooling Tower. Because of the location of the building and the absence of nearby tall buildings, the cooling tower was located in the rear and on the first floor level.

Refrigeration. Two duplicate units of 225 hp size each are installed using dichlorodifluoromethane as the refrigerant. Each compressor is supplied

TABLE 1—TABULATION OF AIR CONDITIONED BUILDINGS

DESIGNATION	LOCATION	NO. OF FLOORS	SQ FT AIR COND. SPACE
Building A	Houston, Tex.	8	169,712
Building B	Annex Bldg. to A, Houston, Tex.	15	80,960
Building C	Boston, Mass.	9	45,000
Building D	Boston, Mass.	11	97,700
Building E	St. Louis, Mo.	19	83,248
Building F	Fresno, Calif.	5	54,100
Building G	Cincinnati, Ohio	15	60,910
Building H	St. Louis, Mo.	12	106,897
Building I	Hershey, Pa.	4	168,128
Building J	Wilmington, Del.	14	190,510

with reduced capacity operating devices to permit power savings in direct proportion to reductions in load.

Heating of indirect type is furnished to supplement existing direct radiation and to furnish tempered, humidified, outside air during winter operation.

Auxiliary. Pneumatic temperature and humidity controls of conventional type, and throw-away type air filters are used in the design.

A compilation of technical data arranged by floors for this particular building is given in Table 2.

BUILDING B—HOUSTON, TEX.

Type of Building. An addition or annex 15 stories in height was added to the original Building A and provisions for complete air conditioning were made in the plans and specifications as prepared. The occupancy was similar to that of Building A, but greater population density was made possible with the incorporation of air conditioning in the original design.

Zoning of the Annex Building B presented a difficult problem due to the unusual exposure. The south side of the building was shaded by Building

TABLE 2—TECHNICAL DATA BY FLOORS, BUILDING A, HOUSTON, TEXAS

	1ST FLOOR	2ND FLOOR	3RD FLOOR	4TH FLOOR	5TH FLOOR	6TH FLOOR	7TH FLOOR	8TH FLOOR	9TH FLOOR	TOTALS
Floor Area, Sq Ft		21,214	21,214	21,214	21,214	21,214	21,214	21,214	21,214	169,712
Ceiling Height, Ft		12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	...
Gross Room Volume, Cu Ft		272,200	272,200	272,200	272,200	272,200	272,200	272,200	254,500	2,159,900
Number of People		230	110	120	115	145	225	160	90	1,195
Floor Area per Person, Sq Ft		92.5	193	177	185	146	94	133	236	142.0
Watts of Light		5,050	3,800	4,280	5,750	4,350	4,600	5,100	4,400	37,330
Avg Watts per Sq Ft Floor Area		0.24	0.18	0.20	0.27	0.20	0.22	0.24	0.21	0.22
Min. Outside Air, Cfm	Not	4,040	2,500	2,540	2,640	2,680	2,840	2,760	5,000	25,000
Min. Outside Air, Cfm per Person	Condi-	17.5	22.7	21.2	23.0	18.5	12.6	17.2	55.5	20.9
Min. Outside Air, Changes per Hour	tioned	0.89	0.55	0.56	0.58	0.59	0.62	0.61	1.18	0.70
Internal Sensible Tons		42.90	28.30	28.80	33.15	30.90	34.25	31.80	34.60	264.0
Internal Latent Tons		10.75	8.25	8.70	13.65	9.60	12.30	9.95	3.45	77.3
Total Internal Tons		53.65	36.55	37.50	46.80	40.50	46.55	41.75	38.05	341.3
Total Tons ÷ Sens. Tons = Int. Mult.		1.25	1.29	1.30	1.41	1.31	1.36	1.31	1.10	1.29
Dew-point Required, deg F		56.8	56	55.8	53.25	55.6	54.5	55.6	59.0	56
Cfm from Dehumidifier		20,250	12,840	13,000	13,470	13,750	14,550	14,140	18,000	120,000
Circulated Air, Cfm		20,250	12,840	13,000	13,470	13,750	14,550	14,140	18,000	120,000
Cfm per Sq Ft Floor Area		0.95	0.60	0.61	0.63	0.65	0.68	0.67	0.85	0.71
Dry-bulb Diffusion, Avg, deg F		23.2	24	24.2	26.7	24.4	25.5	24.4	21	24
Circulated Air Change, Min		13.4	21.2	21.0	20.2	19.8	18.7	19.3	14.1	17.9
Tons Refrigeration Required		71.37	48.44	49.47	59.08	53.02	59.61	54.37	59.32	454.68
Volume per Ton, Cu Ft		3,825	5,640	5,520	4,610	5,140	4,570	5,025	4,300	4,750
Floor Area per Ton, Sq Ft		298	449	431	360	400	356	392	359	374

A for the lower nine floors, the number of separate zones having been selected as follows:

First Floor—2 zones—east and west.
Second Floor—3 zones—north, east and west.
Third–Tenth Floors—3 zones—north, east and west with a supplementary northeast zone added to the north or shaded zone.
Eleventh–Fourteenth Floors—4 zones—north, east, south and west.
Penthouse—4 zones—north, east, south and west.

Apparatus Location. Fan and air dehumidifier equipment of the spray type located in the basement supplies zones one, two and three of floors one to seven. Equipment on the roof supplies zones one, two, three and four of floors 8 to 15 inclusive. Separate supplementary volume control of the air supplied to each zone on every floor is furnished for accurate temperature regulation.

Air Distribution. Furred corridor ceilings are used for separate supply and return air ducts connected to the vertical risers in a ventilation stack provided in the new building.

Cooling Tower. In the tower of the new Annex Building B is located an enclosed concealed forced draft cooling tower of sufficient capacity to supply condensing water for the refrigeration equipment of both Buildings A and B.

Refrigeration. Two water cooling systems of the dichlorodifluoromethane type; one of 400 hp and one of 200 hp size, were furnished and installed in the basement to supply chilled water for dehumidification as required by the two air washers.

Heating. Indirect heating is supplied by the same system of duct work utilized for air distribution during the summer operating season.

Auxiliary. Pneumatic temperature and humidity control and throw-away type air filters are used in the installation.

A tabulation of the technical data for this particular installation is outlined in Table 3.

BUILDING C—BOSTON, MASS.

Type of Building. Existing building of 13 stories is occupied by financial organizations in the Boston business district. It has 5000 sq ft per floor with internal as well as external offices.

Zoning. East and west zones per floor with individual volume control provide for load fluctuations on each of the relatively small floors.

Apparatus Location. The roof is utilized for the location of not only the spray-type air dehumidifier and main supply fan, but for the evaporative condensers of the refrigeration plant as well. A 175 hp reciprocating compressor and shell and tube type water cooler are located in the basement.

Air Distribution. Air distributing duct work is located entirely within the building with supply and return risers.

Cooling Tower. Combined forced draft cooling tower and refrigerant condenser (evaporative type) is utilized to reject heat from the conditioned space. If city water at 80 F initial temperature were used, the cooling tower would normally require 1 per cent of 3 gpm per ton for make-up.

Refrigeration. One 175 hp water cooling unit is installed for the upper

TABLE 3—TECHNICAL DATA BY FLOORS, BUILDING B, HOUSTON, TEXAS

	1ST FLOOR	2ND FLOOR	3RD FLOOR	4TH FLOOR	5TH FLOOR	6TH FLOOR	7TH FLOOR	8TH FLOOR	9TH FLOOR	10TH FLOOR	11TH FLOOR	12TH FLOOR	13TH FLOOR	14TH FLOOR	1ST PENT-HOUSE	TOTALS
Floor Area, Sq Ft	7,850	6,160	5,620	6,500	6,240	7,040	6,630	6,280	5,440	6,400	3,920	3,800	3,760	3,300	2,020	80,960
Ceiling Height, Ft	16.0	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	13.83	15.83	14.83	...
Gross Room Volume, Cu Ft	125,570	79,310	72,305	83,618	80,025	90,470	85,100	80,610	69,970	82,250	50,418	48,875	52,060	49,500	28,290	1,078,371
Number of People	46	70	136	86	89	89	89	89	89	66	49	52	179	106	31	1,266
Floor Area per Person, Sq Ft	170	88	41.4	75.5	70	79	74.5	70.5	61.0	97	80	73	21	31.1	65.1	64.0
Watts of Light	7,000	6,050	7,030	7,200	6,250	6,250	6,250	6,250	6,250	6,620	4,300	3,950	3,800	3,900	3,150	84,250
Avg Watts per Sq Ft Floor Area	0.89	0.97	1.25	1.10	1.0	0.89	0.94	0.99	1.15	1.03	1.1	1.04	1.01	1.18	1.56	1.04
Min. Outside Air, Cfm	5,670	2,740	2,740	2,665	2,630	2,630	2,630	2,630	2,630	4,740	1,885	1,800	2,260	2,140	1,815	41,605
Min. Outside Air, Cfm per Person	123	39.2	20.2	31	29.6	29.6	29.6	29.6	29.6	71.6	38.4	34.6	12.6	20.2	58.5	32.9
Min. Outside Air, Changes per Hour	2.7	2.1	2.3	1.9	1.97	1.75	1.85	1.95	2.26	3.4	2.2	2.2	2.6	2.6	3.8	2.3
Internal Sensible Tons	23.60	11.40	11.40	11.10	10.93	10.93	10.93	10.93	10.93	19.70	7.85	7.50	9.44	8.92	7.58	173.14
Internal Latent Tons	0.59	0.89	1.73	1.10	1.13	1.13	1.13	1.13	1.13	0.84	0.62	0.66	2.28	1.35	0.39	16.10
Total Internal Tons	24.19	12.29	13.13	12.20	12.06	12.06	12.06	12.06	12.06	20.54	8.47	8.16	11.72	10.27	7.97	189.24
Total Tons ÷ Sens. Tons = Int. Mult.	1.02	1.08	1.15	1.10	1.10	1.10	1.10	1.10	1.10	1.04	1.08	1.09	1.25	1.15	1.05	1.09
Dew-point Required, deg F	60	59.3	58.4	59	59	59	59	59	59	60	59.3	59.1	57	58.4	59.7	59.1
Cfm from Dehumidifier	13,600	6,350	6,100	6,100	6,030	6,030	6,030	6,030	6,030	11,400	4,380	4,150	4,350	4,780	4,300	95,660
Circulated Air, Cfm	17,514	7,607	6,863	6,531	6,861	6,790	7,161	6,628	7,362	11,516	4,942	4,885	6,352	4,867	4,300	110,179
Cfm per Sq Ft of Floor Area	2.24	1.23	1.22	1.00	1.10	0.96	1.08	1.05	1.35	1.80	1.26	1.28	1.69	1.47	2.13	1.36
Dry-bulb Diffusion, Avg, deg F	15.5	17.3	19.2	19.7	18.4	18.6	17.6	19.2	17.1	19.8	18.4	17.6	15.7	21.3	20.3	18.0
Circulated Air Change, Min	7.15	10.4	10.5	12.8	11.7	13.3	11.9	12.2	9.5	7.1	10.2	10.0	8.2	10.1	6.6	9.7
Tons Refrigeration Required	47.7	24.6	25.3	24.2	23.9	23.9	23.9	23.9	23.9	40.1	17.4	16.9	21.1	20.1	16.6	373.5
Volume per Ton, Cu Ft	2,630	3,220	2,855	3,450	3,350	3,780	3,560	3,375	2,920	2,040	2,890	2,880	2,470	2,460	1,700	2,880
Floor Area per Ton, Sq Ft	164	250	222	268	260	294	278	262	228	268	225	225	178	164	122	216

TABLE 4—TECHNICAL DATA BY FLOORS, BUILDING C, BOSTON, MASS.

	4TH FLOOR	5TH FLOOR	6TH FLOOR	7TH FLOOR	8TH FLOOR	9TH FLOOR	10TH FLOOR	11TH FLOOR	12TH FLOOR	TOTALS
Floor Area, Sq Ft	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	45,000
Ceiling Height, Ft	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.0	...
Gross Room Volume, Cu Ft	47,500	47,500	47,500	47,500	47,500	47,500	47,500	47,500	45,000	425,000
Number of People	50	50	50	50	50	50	50	100	200	650
Floor Area per Person, Sq Ft	100	100	100	100	100	100	100	50	25	69.2
Watts of Light	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	45,000
Avg Watts per Sq Ft of Floor Area	1	1	1	1	1	1	1	1	1	1.0
Min. Outside Air, Cfm	1,200	1,020	1,105	1,020	1,130	1,105	1,120	1,140	1,660	10,500
Min. Outside Air, Cfm per Person	24	20.4	22.1	20.4	22.6	22.1	22.4	11.4	8.3	16.2
Min. Outside Air, Changes per Hour	1.52	1.29	1.40	1.29	1.43	1.40	1.41	1.44	2.21	1.48
Internal Sensible Tons	6.08	4.22	4.22	4.22	4.72	5.21	7.10	7.91	15.75	59.4
Internal Latent Tons	0.64	0.64	0.64	0.64	0.64	0.64	0.64	1.27	2.55	8.3
Total Internal Tons	6.72	4.86	4.86	4.86	5.36	5.85	7.74	9.18	18.30	67.7
Total Tons ÷ Sens. Tons = Int. Multiplier	1.103	1.15	1.15	1.15	1.135	1.125	1.09	1.16	1.16	1.14
Dew-point Required, deg F	59	58.4	58.4	58.4	58.6	58.7	59.1	58.3	58.3	58.5
Cfm from Dehumidifier	3,350	2,820	3,045	2,820	3,105	3,050	3,090	3,140	4,580	30,500
Circulated Air, Cfm	4,390	3,685	4,000	3,685	4,075	3,990	4,050	4,125	6,000	38,000
Cfm per Sq Ft of Floor Area	0.88	0.74	0.80	0.74	0.82	0.80	0.81	0.83	1.12	0.85
Dry-bulb Diffusion, Avg deg F	15.6	12.9	11.9	12.9	13.0	14.7	19.7	21.6	29.6	16.9
Circulated Air Change, Min	10.8	12.9	11.9	12.9	11.6	11.9	11.7	11.5	7.50	11.2
Tons Refrigeration Required	9.50	7.22	7.42	7.22	7.98	8.41	10.34	11.82	22.14	92.1
Volume per Ton, Cu Ft	5,000	6,580	6,400	6,580	5,950	5,650	4,600	4,010	2,040	4,620
Floor Area per Ton, Sq Ft	526	693	675	693	626	595	484	423	226	488

nine floors, and an additional unit is to be furnished later for the lower three floors.

Heating. Direct radiation already installed was used, and supplemented with outside air tempering at the central supply fan.

Auxiliary. The automatic differential type pneumatic temperature and humidity control is readjusted to maintain a predetermined set of indoor temperatures for various outdoor conditions. An automatic by-pass control was added at the central air dehumidifier to compensate for changes in temperature of return air from the conditioned spaces. Throw-away type air filters were furnished for air cleansing.

In Table 4 are given condensed data applicable to this building.

BUILDING D—BOSTON, MASS.

Type of Building. An existing office building offers space to prospective tenants and was only partially occupied at the time of the remodeling operation which included the installation of air conditioning.

Zoning. The north and the south wings are separately zoned. Individual field assembled surface cooling units are installed in each zone of the building, each unit supplying three floors.

Apparatus Location. Field assembled, surface cooling units located in the north wing of floors 2, 5, 8, and roof, supply the north zone of the following floors respectively. The second floor north unit supplies the north zone of floors 1 to 3. The fifth floor north unit supplies floors 4 to 5 north zone. The eight floor north unit supplies the north zone of floors 7 to 9. The north unit on the roof supplies the north zone of the tenth and eleventh floors. Similar units located on the same floors supply the south zone of identical floors in the south wing of the building.

Air Distribution. Duct work is installed within the office space rather than through the corridors and is furred to match the existing surroundings as indicated in Fig. 1.

The supply air is directed toward the outside walls of the room in each case.

Cooling Tower. A forced draft cooling tower is located on the roof to reject the heat removed from the building to the outside air by means of condensing water circulated through the refrigerant condensers in the basement. The cooling tower was located so that it was not visible from the street and surrounding buildings which are relatively lower in height; consequently its operation is not considered objectionable.

Refrigeration. A 250-hp water cooling system with two 125-hp compressor units and a single water cooler was located in the basement and functions to furnish the chilled water required for the surface cooling units located on the upper floors.

Heating. Outside air tempering is furnished through the heating coils in the field assembled air conditioning units located on the various floors, winter dehumidification being supplied through the use of atomizing sprays. The existing direct radiation in the building was allowed to remain and functions as previously installed.

Auxiliary. A differential system of automatic temperature and humidity control is furnished to automatically regulate the indoor temperature in ac-

cordance with the outside temperature utilizing automatic by-passing of recirculated air around the finned cooling surface of each air conditioner to maintain a uniform condition within the conditioned space during summer operation. Winter heating is also automatically controlled by a system of pneumatic control instruments.

Air filtering is furnished through the use of cleanable type air filters assembled in the field in conjunction with the fan and surface cooling and heating equipment.

Detailed figures are given in Table 5 for the unit factors applicable to Building D.

BUILDING E—ST. LOUIS, MO.

Type of Building. At the time of installation of air conditioning in this 20-story building in St. Louis, a portion of the space was not rented, gov-

FIG. 1. FINISHED DUCT APPEARANCE IN BUILDING D

erning to some degree the selection of unitary surface cooling apparatus on each floor for providing air conditioning. Duct work is installed in all of the occupied zones, and on the unoccupied floors the air conditioning units only, have been installed and connected to the refrigerating equipment. The installation on these floors is to be completed at such time as this space is rented, duct work then being located to suit the partition arrangement required by the tenant.

Zoning. On the lower floors, two air conditioning units are installed per floor, whereas, on the upper floors a single conditioner of adequate capacity is furnished. In general the zoning is east and west per floor with the individual units being automatically controlled for the independent control of temperature and humidity on each floor and in each zone.

Apparatus Location. Unit air conditioners of the factory assembled type, as illustrated by Fig. 2, are installed throughout the building, two units being furnished on the lower floors, and one unit on each of the upper floors. A

central water cooling plant is located in the basement, with chilled water being supplied to the air conditioners for cooling purposes.

Air Distribution. Duct work is installed along the faces of the columns within the building and furred in without making an effort to balance the furred space on both sides of the column. The finished appearance is illustrated by Fig. 3.

Cooling Tower. A forced draft cooling tower is used for rejecting the heat removed from the conditioned space to the outside air.

Refrigeration. Refrigeration for dehumidification is furnished by a two compressor dichlorodifluoromethane water cooling system with one large water

TABLE 5—TECHNICAL DATA BY FLOORS, BUILDING D, BOSTON, MASS.

	1ST FLOOR	2ND FLOOR	3RD TO 10TH FLOORS INC. EACH FLOOR	11TH FLOOR	TOTALS
Floor Area, Sq Ft	10,600	10,600	8,500	8,500	97,700
Ceiling Height, Ft	9.5	8.25	8.25	8.25	....
Gross Room Volume, Cu Ft	100,000	87,500	70,000	70,000	817,500
Number of People	120	120	120	120	1,320
Floor Area per Person, Sq Ft	88.4	88.4	70.8	70.8	74.0
Watts of Light	7,000	3,500	3,500	3,500	42,000
Avg Watts per Sq Ft of Floor Area	0.66	0.33	0.41	0.41	0.43
Min. Outside Air, Cfm	2,200	2,200	2,200	2,200	24,200
Min. Outside Air, Cfm per Person	18.3	18.3	18.3	18.3	18.3
Min. Outside Air Changes per Hour	1.32	1.51	1.89	1.89	1.78
Internal Sensible Tons	7.81	8.29	8.29	16.0	98.4
Internal Latent Tons	1.53	1.53	1.53	1.53	16.8
Total Internal Tons	9.34	9.82	9.82	17.53	115.2
Total Tons + Sens. Tons = Int. Mult.	1.19	1.18	1.18	1.10	1.17
Dew-point Required, deg F	59.0	59	59	59.8	59.1
Cfm from Dehumidifier	7,000	7,000	7,000	12,300	82,300
Circulated Air, Cfm	7,000	7,000	7,000	12,300	82,300
Cfm per Sq Ft of Floor Area	0.66	0.66	0.82	1.45	0.84
Dry-bulb Diffusion, Avg deg F	12.5	13.2	13.2	14.5	13.4
Circulated Air Change, Min	14.3	12.5	10.0	5.70	9.9
Tons Refrigeration Required	14.4	14.8	14.8	22.5	170.1
Volume per Ton, Cu Ft	6,950	5,900	4,730	3,110	4,800
Floor Area, per Ton, Sq Ft	736	715	574	378	574

cooler operating in conjunction with a 200 hp and a 125 hp machine. This provides considerable flexibility in operation for the many and varied load factors involved in office building air conditioning.

Heating. The direct radiation in the building is supplemented with the tempering of outside air through the use of heating coils in the air conditioning units.

Auxiliary. A pneumatic system of temperature and humidity control is used in the building for year around temperature control.

Throw-away type air filters are furnished in the base of each vertical floor type air conditioner and are indicated clearly on Fig. 2.

A compilation of technical data arranged by floors for this particular building is given in Table 6.

BUILDING F—FRESNO, CALIF.

Type of Building. In the exceedingly hot, dry climate of Fresno, Calif., the use of evaporative cooling was originally deemed practical and such an installation was made by the owners. Later a mechanical refrigeration system designed along special lines was used to replace the original adiabatic cooling system.

There is a multiplicity of private offices arranged along the corridor of the property.

Zoning. The building is zoned principally into two zones per floor in the

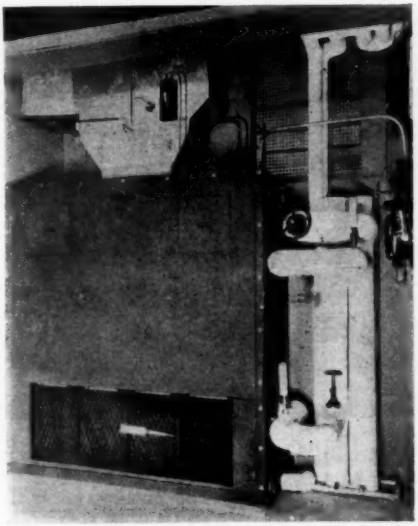

FIG. 2. FLOOR TYPE CONDITIONER IN BUILDING E

east and west wings, the building limitations deciding this factor to a large degree.

Apparatus Location. Surface coils for cooling the air supplied to the conditioned spaces are installed in the outlet of the existing adiabatic air washer. The refrigeration equipment is separated, with the evaporator coils of the direct expansion type being located on the inlet end of the main supply fan in the basement near the compressor, while the condenser is installed on the roof under the cooling tower.

Air Distribution. As the simplest system of air distribution was to be devised for this particular installation, Fig. 4 shows the method of applying building board below the existing furred ceiling, forming a plenum space used for distributing the conditioned air from the main supply fan. Openings were cut from the corridor at the ceiling line into each office above the entrance doors, with supply grilles being located as indicated on Fig. 5.

The transoms above the entrance doors were returned to the factory in which they were made, so that their height could be reduced a sufficient amount to allow for the installation of the supply grilles as indicated. The building-board which forms the bottom of the air supply duct and the corridor ceiling is plastered on the corridor side and the upper side is lined with a 1/16-in. asbestos sheet.

Cooling Tower. The cooling tower furnished for this installation serves the double purpose of furnishing water which is used to precool outside air prior to the cooling of this air by mechanical refrigeration. In addition the water leaving the outside air precooling coil is passed through the con-

FIG. 3. DUCT ARRANGEMENT IN BUILDING E

denser of the refrigeration system. The cooling tower furnished is of the atmospheric type.

Refrigeration. Because such a large percentage of the heat to be removed from the building is in the form of sensible heat at a high temperature level, the use of cooling tower water for precooling reduces the refrigerating capacity necessitated for this installation to one-half that which would normally be required. It is also possible to make use of 100 per cent outside air and thus raise the ventilation standard within the interior without a material penalty in operating cost. The 75 hp refrigeration machine with reduced capacity control is arranged to operate with direct expansion evaporator coils located in the basement.

Heating. The air conditioning equipment was furnished without provisions for operating during the winter.

BUILDING G—CINCINNATI, OHIO

Type of Building. A 15-story office building, consisting of 14 floors, was rented to individual tenants and one floor was used for office space by the owner. The lower 12 floors have an overall dimension of approximately

TABLE 6—TECHNICAL DATA BY FLOORS, BUILDING E, ST. LOUIS, MO.

	1ST FLOOR	2ND FLOOR	3RD FLOOR	4TH FLOOR	5TH FLOOR	6TH FLOOR	7TH TO 15TH FLOOR INC. EACH FLOOR	16TH FLOOR	17TH FLOOR	18TH FLOOR	19TH FLOOR	20TH FLOOR	TOTALS
Floor Area, Sq Ft.	575		9,212	9,133	9,144	3,720	3,954	3,808	2,737	3,111	3,111	3,111	83,248
Ceiling Height, Ft.	13.0		11.83	11.83	11.83	10.83	10.83	10.83	10.83	10.83	10.83	10.83	
Gross Room Volume, Cu Ft.	7,475		109,000	108,000	107,000	40,400	42,850	42,400	29,620	34,750	34,750	34,750	933,795
Number of People.	5		150	150	150	20	20	20	20	20	20	20	755
Floor Area per Person, Sq Ft.	115		61.5	60.7	60.8	186	197	190	136	155	155	155	110.2
Watts of Light	800		22,000	22,000	22,000	2,000	2,000	2,000	2,000	2,000	2,000	2,000	96,800
Avg Watts per Sq Ft Floor Area.	1.39	Air Condi-tioning Units In-stalled. Duct Fur-nished at Later Date to Suit Tenants	2.38	2.41	2.4	0.54	0.51	0.52	0.73	0.64	0.64	0.64	1.16
Min. Outside Air, Cfm	200		2,000	2,000	1,700	800	800	700	800	800	800	800	17,800
Min. Outside Air, Cfm per Person	40		13.3	13.3	11.3	40	40	40	40	40	40	40	23.6
Min. Outside Air, Changes per Hour	1.61		1.1	1.1	0.95	1.19	1.12	0.99	1.62	1.38	1.38	1.38	1.14
Internal Sensible Tons	1.75		16.04	16.04	13.06	6.72	6.72	5.55	5.33	5.33	5.33	5.33	141.0
Internal Latent Tons	0.41		2.88	2.88	1.79	1.20	1.20	0.94	0.94	0.94	0.94	0.94	24.7
Total Internal Tons	2.16		18.92	18.92	14.85	7.92	7.92	6.49	6.27	6.27	6.27	6.27	165.7
Total Tons + Sens. Tons = Int. Mult.	1.23		1.18	1.18	1.14	1.18	1.18	1.17	1.18	1.18	1.18	1.18	1.18
Dew-point Required, deg F.	58.5		59	59	58.8	59.5	59.5	58.8	59	59	59	59	59.5
Cfm from Dehumidifier	1,200		12,000	12,000	8,000	4,000	4,000	4,000	4,000	4,000	4,000	4,000	93,200
Circulated Air, Cfm.	1,200		12,000	12,000	8,000	4,000	4,000	4,000	4,000	4,000	4,000	4,000	93,200
Cfm per Sq Ft of Floor Area	2.09		1.3	1.3	0.87	1.07	1.01	1.05	1.46	1.28	1.28	1.28	1.12
Dry-bulb Diffusion, Avg, deg F.	57.2		15.7	15.7	19	19.4	19.4	16	17.7	17.7	17.7	17.7	13.0
Circulated Air Change, Min.	6.22		9.1	9	13.4	10.1	10.7	10.6	7.4	8.7	8.7	8.7	10.0
Tons Refrigeration Required	2.77		24.4	24.4	19.5	10.1	10.1	8.4	9.25	9.25	9.25	9.25	217.4
Volume per Ton, Cu Ft.	2,760		4,460	4,440	5,500	4,000	4,240	5,050	3,200	3,750	3,750	3,750	4,290
Floor Area per Ton, Sq Ft.	213		378	374	468	368	391	454	296	336	336	336	383

100 ft x 60 ft, the upper three floors form a tower, roughly 60 ft x 60 ft.

Zoning. Building is zoned east and west with separate recirculating fans on each floor receiving a supply of conditioned air from the central station dehumidifiers.

Apparatus Location. Two dehumidifiers of the spray type are installed, one being located on the fourth floor to take care of floors 1 to 5, the other being located on the tenth floor in an equipment room. This latter dehumidifier supplies air to recirculating fans on each of the upper ten floors. A central water cooling system is located in the basement, chilled water being supplied to each of the two dehumidifiers under automatic control in accordance with heat gain requirements.

Local recirculating fans are used for the east and west zones, two fans per floor; they are located in the center stair wells adjacent to elevator shaft.

Air Distribution. On this installation, an unusual type of system is utilized for the distribution of conditioned air over the space. The supply ducts were arranged along the outside building wall, with air being discharged towards the center of the building. The duct work is installed to appear as false beams, with square elbows being utilized throughout.

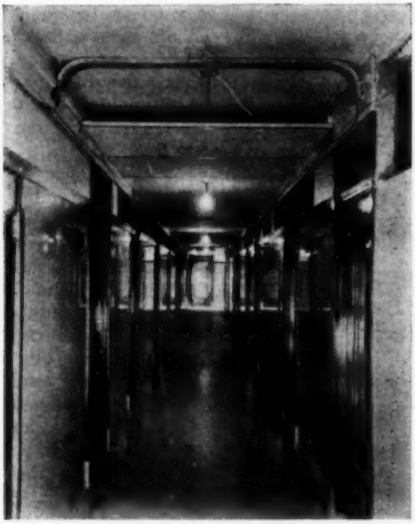

FIG. 4. PLENUM DUCT CONSTRUCTION IN BUILDING F

The duct work was reduced in size on the side of the outside wall, the space between the wall and the bottom of the duct being filled by a sheet metal separation.

Unique in the method of duct manufacture is the fact that the sheet metal joints were welded to appear as indicated in Fig. 6. The joints were then filled and two coats of paint were applied directly to the sheet metal without the use of furring or a plaster finish. The finished installation is shown by Fig. 7 which can be compared with the original building appearance prior to the installation as indicated on Fig. 8.

City water is utilized for condensing purposes under automatic control of a gas pressure actuated water regulating valve, the quantity required being controlled in step with the automatic regulation of the refrigeration capacity.

Refrigeration. A 210-hp central station water cooling system was furnished with one 150-hp manually controlled compressor and one 60-hp automatic compressor. Each of the refrigerating machines is itself furnished with capacity reducing by-pass valves so that operation from 15 to 100 per cent capacity can be obtained with proportionate reduction in power input to the compressor motors for reduction in heat gain within the space.

FIG. 5. LOCATION OF SUPPLY GRILLE ABOVE DOOR IN BUILDING F

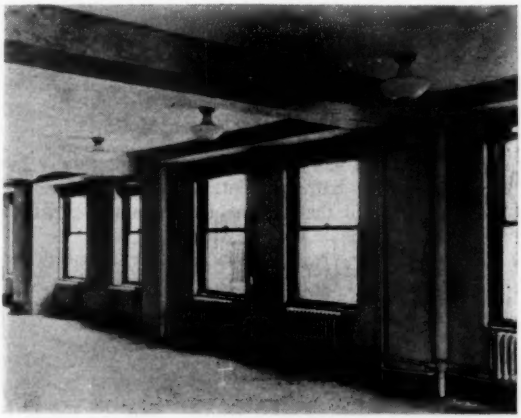

FIG. 6. WELDED DUCT INSTALLED IN BUILDING G

Heating. Heating of the individual office space is obtained by the use of direct radiation. The air conditioning equipment is operated during the winter time to supply to the conditioned space tempered, humidified outside air for ventilation purposes.

Auxiliary. Differential thermostatic control of the pneumatic type is installed, with the summer indoor temperature automatically adjusted in accordance with outdoor temperature. Localized zone control is furnished by automatic damper regulation of the quantity of dehumidified air supplied to each of the local recirculating fans in the separate zones.

FIG. 7. DUCT WORK PAINTED IN BUILDING G

Throw-away type filters are furnished for supplementary cleaning of the air supply to the conditioned space.

In Table 7 are given complete technical data arranged by floors for this particular building.

BUILDING H—ST. LOUIS, MO.

Type of Building. This 12-story building has an unusually large southern exposure with an exceedingly large percentage of the wall being glass. Sun effect or the heat gain due to the afternoon sun is the greatest single item

FIG. 8. ROOM APPEARANCE BEFORE DUCT INSTALLATION IN BUILDING G

of the total load and was, therefore, one of the governing factors of the design.

Zoning. Each floor of the building is designed with a separate recirculating fan. Supplementary control is furnished for four independent zones including a west sun zone, an east sun zone, a west shaded zone and an east shaded zone.

Apparatus Location. The layout of equipment utilized on this project required the use of no office floor space. Local recirculating fans for each floor are mounted from the ceiling.

The refrigeration equipment is located in the basement. In the space directly above, and in a court to the rear and outside, was built an air washer room with vertical supply and return duct risers located outside of the building.

Air Distribution. Duct work had to be run internally within the conditioned space and fitted around existing cross-beams and girders which, due to the shape of the building, interfered to a considerable degree. The finished duct work as furred-in with sound absorbing material is indicated in the

TABLE 7—TECHNICAL DATA BY FLOORS, BUILDING G, CINCINNATI, OHIO

	1ST FLOOR	2ND FLOOR	3RD AND 4TH FLOORS EACH FLOOR	5TH FLOOR	6TH TO 8TH FLOORS INC. EACH FLOOR	9TH TO 11TH FLOORS INC. EACH FLOOR	12TH FLOOR	13TH FLOOR	14TH FLOOR	15TH FLOOR	TOTALS
Floor Area, Sq Ft	2,890	2,890	2,890	4,940	4,940	4,940	4,940	3,695	3,025	3,110	60,910
Ceiling Height, Ft	16.0	11.0	11.0	10.5	10.5	10.5	10.5	10.5	10.5	9.5	...
Gross Room Volume, Cu Ft	46,300	32,800	32,800	51,500	51,500	51,500	51,500	38,800	31,800	35,240	662,540
Number of People	30	50	30	60	45	45	45	35	35	50	635
Floor Area per Person, Sq Ft	96.3	57.8	96.3	82.3	109.5	109.5	109.5	105.2	86.5	62.2	95.8
Watts of Light	5,150	6,000	6,000	8,800	8,800	8,800	8,800	5,650	5,400	5,400	110,000
Avg Watts per Sq Ft Floor Area	1.78	2.08	2.08	1.78	1.78	1.78	1.78	1.53	1.78	1.73	1.81
Min. Outside Air, Cfm	1,085	770	770	1,205	1,205	1,205	1,205	906	743	825	15,509
Min. Outside Air, per Person, Cfm	36.16	15.4	25.6	20	26.8	26.8	26.8	25.9	21.2	16.5	24.4
Min. Outside Air, Changes per Hour	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.41
Internal Sensible Tons	6.47	5.02	4.63	7.47	6.68	6.98	7.82	5.60	4.92	7.11	94.0
Internal Latent Tons	1.15	1.19	0.93	1.62	1.44	1.44	1.44	1.10	0.98	1.02	19.0
Total Internal Tons	7.62	6.21	5.56	9.09	8.12	8.42	9.26	6.70	5.90	8.13	113.0
Total Tons ÷ Sens. Tons = Int. Mult.	1.18	1.24	1.20	1.21	1.22	1.21	1.18	1.19	1.20	1.14	1.20
Dew-point Required, deg F	58	57	57.67	57.5	57.3	57.5	58	57.8	57.6	58.5	57.7
Cfm from Dehumidifier	3,250	2,415	2,285	3,670	3,400	3,580	4,100	2,910	2,540	3,825	46,600
Circulated Air, Cfm	4,320	3,240	2,960	4,890	4,326	4,530	5,380	3,820	3,200	5,184	62,522
Cfm per Sq Ft of Floor Area	1.50	1.12	1.02	0.99	0.88	0.92	1.09	1.03	1.06	1.66	1.03
Dry-bulb Diffusion Avg, deg F	16.6	17.1	17.3	17.0	17.9	17.7	16.7	16.9	17.8	15.9	17.3
Circulated Air Change, Min	10.7	10.1	11.1	10.5	11.9	11.4	9.60	10.1	10.0	6.80	10.6
Tons Refrigeration Required	11.36	8.97	8.30	13.12	12.56	12.82	13.61	10.03	8.72	11.26	169.8
Volume per Ton, Cu Ft	4,070	3,660	2,960	3,920	4,100	4,000	3,780	3,860	3,650	3,140	3,900
Floor Area per Ton, Sq Ft	254	322	260	376	393	384	362	368	347	276	359

photograph of the office space shown in Fig. 9. The location of the duct work in relation to the building columns is indicated in Fig. 9.

Cooling Tower. A cooling tower of the forced draft type is installed for cooling the condensing water utilized by the refrigeration plant.

Refrigeration. The dichlorodifluoromethane water cooling system furnished for this project is of 400-hp capacity and divided into two compressor units. Each 200-hp unit is operated in conjunction with a single water cooler of the shell and tube type to provide chilled water for dehumidification during summer operation of the equipment.

Heating. Direct radiation was left in place and supplementary heating furnished by tempering outside air during winter operation.

Auxiliary. Automatic controls of the pneumatic type are provided. The recirculating fans located on each floor are furnished with supplementary

FIG. 9. DUCT INSTALLATION IN BUILDING H

automatic volume controls for the four zones on each floor to provide for further variation in heat gain.

Throw-away type filters are furnished for air cleaning to supplement the action of the spray type air dehumidifier.

Included in Table 8 are complete technical data arranged by floors for Building H.

BUILDING I—HERSHEY, PA.

Type of Building. A complete 4-story windowless office building, shown in Fig. 10, is used as the home office building of a large manufacturer. The walls are insulated and a water level is maintained on the roof to reduce the heat gain added by the sun's rays during summer operation.

Zoning. With insulated walls and the omission of windows, the needs for zoning are dependent almost entirely upon internal occupancy and internal loads. The basement and first floor are supplied by one air condition-

ing system, and the second and third floors by a second air conditioning system.

Apparatus Location. All of the mechanical equipment for refrigeration and air conditioning is located in the basement, with duct work being furnished to supply and recirculate the air to the several spaces.

Air Distribution. Various types of air distribution are combined in the several floors of the building; some duct work being exposed where its appearance is not objectionable, while some is installed to appear as false beams as illustrated in Fig. 11.

TABLE 8—TECHNICAL DATA BY FLOORS, BUILDING H, ST. LOUIS, MO.

	1ST FLOOR	2ND, 4TH 8TH, 10TH AND 11TH FLOORS EACH FLOOR	3RD, 5TH 6TH AND 7TH FLOORS EACH FLOOR	12TH FLOOR	FLOOR 12-A	TOTALS
Floor Area, Sq Ft	8,401	8,938	8,081	8,413	4,075	106,897
Ceiling Height, Ft	16.75	10.0	10.0	10.0	9.0	....
Gross Room Volume, Cu Ft	154,500	89,380	89,380	84,130	36,675	1,169.105
Number of People	120	120	120	120	75	1,515
Floor Area per Person, Sq Ft	70	74	67	70	54	70.5
Watts of Light	17,890	17,890	17,890	15,950	8,211	223,041
Avg Watts per Sq Ft Floor Area	2.13	2.00	2.22	1.90	1.96	2.09
Min. Outside Air, Cfm	3,870	2,215	2,215	2,100	920	29,040
Min. Outside Air, Cfm per Person	32.2	18.6	18.4	17.5	12.3	19.2
Min. Outside Air, Changes per Hour	1.50	1.50	1.50	1.60	1.50	1.49
Internal Sensible Tons	26.9	17.4	17.4	21.6	19.7	242.8
Internal Latent Tons	2.20	2.00	2.00	2.00	1.30	25.5
Total Internal Tons	29.1	19.4	19.4	23.6	21.0	268.3
Total Tons ÷ Sens. Tons = Int. Mult	1.08	1.11	1.11	1.09	1.07	1.11
Dew-point Required, deg F	55.8	55.33	55.33	55.67	56	55.4
Cfm from Dehumidifier	13,650	9,100	9,100	11,150	9,900	108,500
Circulated Air, Cfm	15,200	13,100	13,100	15,800	15,000	177,400
Cfm per Sq Ft of Floor Area	1.81	1.56	1.62	1.88	3.68	1.66
Dry-bulb Diffusion, Avg, deg F	20.2	15.0	15.0	15.6	15.0	15.5
Circulated Air Changes, Min	10.2	6.80	6.80	5.30	2.40	6.6
Tons Refrigeration Required	42.3	26.9	26.9	30.6	23.9	366.4
Volume per Ton, Cu Ft	3,660	3,320	3,320	2,750	1,540	3,180
Floor Area per Ton, Sq Ft	199	332	300	275	170	291

Well water is used for condensing purposes.

Refrigeration. Three 125-hp water cooling systems are installed in the basement to provide chilled water for the spray type dehumidifiers utilized in controlling the temperature and humidity.

Heating. Heating is accomplished entirely by an indirect method. The central station dehumidifiers are provided with tempering and reheat coils which provide the full amount of heat necessary in the conditioned space. This is made possible by the insulation of the walls and the fact that there are no windows or spaces with varying exposure. The wall insulation tends to reduce the amount of heat leakage to a minimum and the lack of windows practically eliminates localized infiltration of outside air.

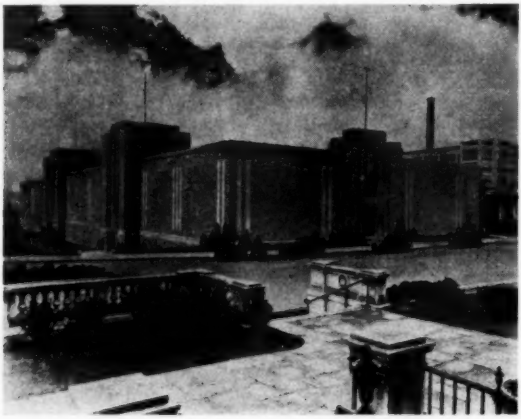

FIG. 10. WINDOWLESS OFFICE BUILDING I

Auxiliary. A differential system of pneumatic temperature and humidity control is provided to automatically adjust the indoor temperature in accordance with changes in outdoor temperature.

Throw-away type air filters are furnished for air cleaning.

Complete technical data for Building I and arranged by floors are given in Table 9.

BUILDING J—WILMINGTON, DEL.

Type of Building. This particular building was newly constructed for occupancy by a single firm with the exception of the store space located in the lower floors which was to be leased to individual tenants.

In a new building of this type it was possible to install a system which did not interfere with the architectural treatment and that chosen in this particular case represents a unique design.

Zoning. Three separate zones were selected for the basement, first and second floors. One system was designed for the north stores, another system air conditioned south stores and a third system took care of the office spaces on the first and second floors.

The remainder of the building was zoned into north and south, with apparatus on the fourteenth floor taking care of the north portion of the third to thirteenth floors, while a separate unit on the fifteenth floor level is provided for the south half of the third to thirteenth floors.

Apparatus Location. An 800 hp water cooling system is located in the basement with three spray-type air dehumidifiers to air condition the space up to and including the second floor. Dehumidifiers for floors 3 to 13 are located on 14 to 15 floor levels, with the south zone being located on the fifteenth floor directly above the north zone unit on the fourteenth floor.

Air Distribution. On the lower floors an overhead plaque system of air

distribution is installed in the conventional manner with duct work being concealed in the furred ceilings above the conditioned space.

On floors 3 to 13, radiator enclosures or air mixing units of the high pressure type are provided of such a design that considerable air from the room is mixed in the summer time with the cold supply air issuing from the orifices in the top of the radiator enclosure so that the temperature of the air stream 3 ft from the supply opening is at least 15 F higher than the temperature of air in the supply duct. A photograph of an actual radiator enclosure is illustrated in Fig. 12, which serves the double function of providing winter heating and also as a means of introducing a supply of conditioned air. The lattice grille opening at the floor line insures recirculation of room air at the floor level during winter operation. The rear openings nearest the window pane are directly above a finned direct heating surface located in the unit. The air is discharged vertically from these openings with room air being withdrawn from the floor line, passed over the heating surface and then out through the openings in the top of the unit to provide a current of warm air directly in front of the window pane.

The front row of openings on the room side of the enclosure and in the top of the unit, conceal oval-shaped air discharge orifices through which cold dehumidified air is discharged in the summer time and through which tempered, humidified, outside air is introduced in the winter time. This front row of openings is the only supply of conditioned air to the room itself for temperature control and can be manually adjusted by a regulator provided in the top of the enclosure. The second control knob serves as a means of adjustment for the self-contained steam regulating valve for winter heating control.

Vertical duct risers from the third floor to the fourteenth floor are run along the outside walls adjacent to the air mixing units located under the windows and connections are made from the vertical duct risers to each unit. The duct risers are arranged along the outside wall adjacent to columns.

Cooling Tower. A forced draft cooling tower is furnished for reclaiming

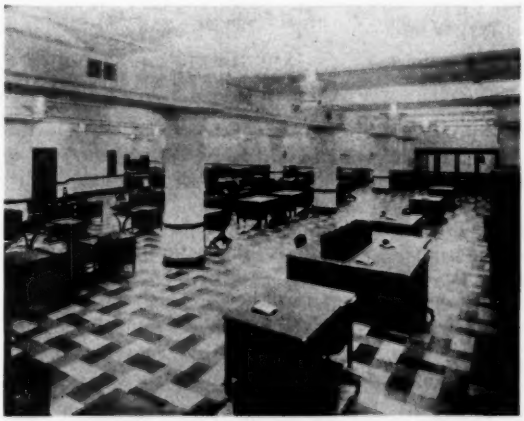

FIG. 11. DUCT ARRANGEMENT IN BUILDING I

the condensing water utilized by the refrigeration system installed for summer dehumidification purposes.

Refrigeration. A water cooling system of 800-hp size with two 400-hp compressors is installed in the basement. Chilled water is circulated from the refrigerating system through heat exchanging surfaces in the spray-type air dehumidifiers. Spare refrigeration capacity is provided for future extension of the air conditioned space.

Heating. Winter heating is obtained by the use of finned heaters concealed within the radiator enclosures which serve as a means for air distribution

TABLE 9—TECHNICAL DATA BY FLOORS, BUILDING I, HERSHEY, PA.

	BASEMENT	1ST FLOOR	2ND FLOOR	3RD FLOOR	TOTALS
Floor Area, Sq Ft	31,500	35,000	51,529	50,099	168,128
Ceiling Height, Ft	10.0	17.66	12.0	14.0	....
Gross Room Volume, Cu Ft	315,000	618,120	621,350	701,400	2,492,106
Number of People	40	1,030	772	173	2,015
Floor Area per Person, Sq Ft	787	34	668	290	83.7
Watts of Light	94,500	94,158	115,536	102,750	406,944
Avg Watts per Sq Ft Floor Area	3.0	2.69	2.24	2.04	2.42
Min. Outside Air, Cfm	3,065	5,315	6,485	3,380	18,245
Min. Outside Air, Cfm per Person	76.6	5.1	8.4	19.5	9.07
Min. Outside Air, Changes per Hour	0.58	0.52	0.62	0.28	0.44
Internal Sensible Tons	96.2	63.5	56.5	88.8	305.0
Internal Latent Tons	0.7	16.9	11.7	2.2	31.5
Total Internal Tons	96.9	80.4	68.2	91.0	336.5
Total Tons ÷ Sens. Tons = Int. Mult.	1.00	1.26	1.21	1.02	1.10
Dew-point Required, deg F	61	61	52	53	60
Cfm from Dehumidifier	44,600	29,400	24,730	43,600	142,330
Circulated Air, Cfm	59,500	43,700	53,700	82,700	239,600
Cfm per Sq Ft of Floor Area	1.89	1.25	1.04	1.65	1.42
Dry-bulb Diffusion, Avg, deg F	18	16	11.5	11.9	11.9
Circulated Air Change, Min	5.3	14.1	11.6	8.5	10.4
Tons Refrigeration Required	112.6	94.3	94.5	110.3	411.7
Volume per Ton, Cu Ft	2,800	6,550	6,590	6,350	6,050
Floor Area per Ton, Sq Ft	280	371	546	455	408
Inside Air Design Conditions	85 F DB 45% RH	85 F DB 50% RH	85 F DB 50% RH	70 F DB 50% RH	....

summer and winter. Dehumidified air tempered to the room temperature is supplied to the air mixing units through the duct work used in the summer time for cooling.

Auxiliary. A differential type of temperature humidity control is utilized in which the indoor temperature is automatically regulated in accordance with the outside street temperature.

Automatic, oil type filters are furnished for air cleaning.

In Table 10 are given complete technical data for Building J arranged by floors.

COMPOSITE SUMMARY OF TECHNICAL DATA

In further explanation of the technical data submitted for each of the foregoing buildings, certain of the items tabulated may be further amplified.

The item *floor area in square feet* includes only the net air conditioned space to which air is supplied from the system.

The *minimum outside air cfm* is that quantity of outside air introduced to the air conditioning system under a condition of maximum load at the design point tabulated in each case. This is the quantity of outside air including infiltration which has been estimated to be introduced to the building to maintain a definite standard of air purity within the enclosure.

The *internal sensible tons* is the internal sensible heat gain within the conditioned space including heat transmission through the walls and glass, heat from electric lighting, sensible heat from people and any other source of dry heat within the enclosure itself, which is a component part of the internal sensible heat gain. The internal sensible load also includes that small

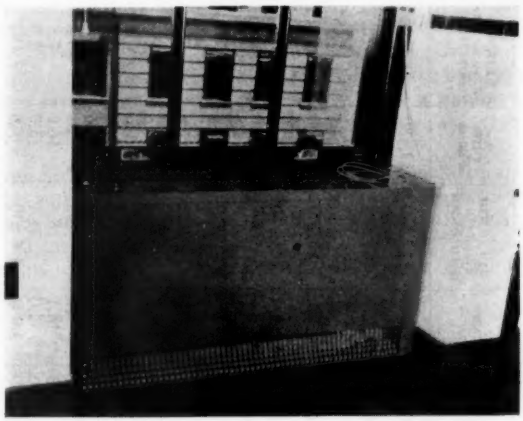

FIG. 12. RADIATOR ENCLOSURE IN BUILDING J

portion of the outside air which infiltrates direct to the conditioned space thereby adding to the internal sensible heat load.

The *internal latent tons* includes the internal latent heat load within the conditioned space including the latent heat of infiltrated outside air exclusive of outside air introduced into the air conditioning apparatus for ventilation purposes.

The term *internal multiplier* which is defined as *total tons divided by sensible tons* is the ratio of total internal heat to internal sensible heat. This relationship is set up to determine the dew-point of the air required to be supplied to the conditioned space for maintaining the predetermined condition of temperature and humidity for the calculated internal, sensible and latent, heat load.

The term *dry-bulb diffusion average* is the temperature rise experienced by the supply air from the time it leaves the supply register until it is withdrawn from the room through the return air grille. This term *diffusion* is also the temperature range through which a specified quantity of circulated

TABLE 10—TECHNICAL DATA BY FLOORS, BUILDING J, WILMINGTON, DEL.

	GROUND FLOOR	1ST FLOOR	2ND FLOOR	3RD TO 9TH FLOOR INC. EACH FLOOR	10TH FLOOR	BETWEEN 10TH AND 11TH FLOORS MEETING ROOM	11TH FLOOR	12TH FLOOR	13TH FLOOR	TOTALS
Floor Area, Sq Ft	14,130	18,800	18,800	13,400	12,180	1,220	13,400	9,700	9,700	190,510
Ceiling Height, Ft	13.5	11.0	11.0	10.58	10.58	16.0	10.58	10.58	10.58	...
Gross Room Volume, Cu Ft	190,600	206,800	206,800	141,800	128,500	19,520	141,800	102,200	102,200	2,071,500
Number of People	354	184	194	134	117	100	117	90	90	2,084
Floor Area per Person, Sq Ft	40	102	97	100	104	122	114	108	108	91.3
Watts Light	42,440	56,400	56,400	40,200	35,100	5,100	35,100	29,100	29,100	565,040
Avg Watts per Sq Ft Floor Area	2.99	3.0	3.00	3.00	2.88	4.18	2.6	3.0	3.0	2.97
Min. Outside Air, Cfm	8,850	4,800	5,200	3,070	2,690	830	3,800	3,040	3,150	53,020
Min. Outside Air, Cfm per Person	25	26	26.8	22.9	23.0	8.3	32.4	33.8	35	25.4
Min. Outside Air, Changes per Hour	2.78	1.39	1.51	1.3	1.25	2.5	1.61	1.78	2.1	1.54
Internal Sensible Tons	36.2	25.5	27.9	18.5	16.2	5.0	22.9	18.3	18.9	295.4
Internal Latent Tons	4.6	3.9	4.3	1.7	1.4	1.3	1.4	1.1	1.1	29.7
Total Internal Tons	40.8	29.4	32.2	20.2	17.6	6.3	24.3	19.4	20.0	325.1
Total Tons ÷ Sens. Tons = Int. Mult.	1.13	1.18	1.18	1.09	1.09	1.26	1.06	1.06	1.06	1.10
Dew-point Required, deg F	58.5	58.0	58	58	59	56.5	57.6	57.6	57.6	59.0
Cfm from Dehumidifier	18,650	13,500	14,700	9,210	8,050	2,500	11,400	9,100	9,480	149,350
Circulated Air, Cfm	18,650	13,500	14,700	9,210	8,050	2,500	11,400	9,100	9,480	149,350
Cfm per Sq Ft of Floor Area	1.32	0.72	0.78	0.69	0.66	2.0	0.85	0.94	0.97	0.78
Dry-bulb Diffusion, Avg, deg F	21.5	22	22	22	21	23.5	22.4	22.4	22.4	21
Circulated Air Change, Min	10.2	15.3	14.1	15.4	16.0	7.8	12.4	11.2	10.8	13.9
Tons Refrigeration Required	65.5	48.0	52.0	31.4	25.5	7.9	39.9	31.8	33.1	515.6
Volume per Ton, Cu Ft	2,910	4,300	3,980	4,500	5,050	2,480	3,550	3,220	3,100	4,020
Floor Area per Ton, Sq Ft	216	392	362	426	476	155	336	305	293	369

air would be warmed in absorbing the internal sensible heat load in the conditioned space.

The *circulated air change in minutes* is the number of minutes required for the complete room volume to be displaced by the supply fan air quantity. This is, therefore, the cubical content of the building divided by the fan air capacity in cubic feet per minute.

The term *tons refrigeration required* is the internal total sensible and latent load plus the load due to outside air introduced for ventilation purposes under maximum heat gain conditions.

The *cubic feet volume per ton* and *square feet floor area per ton* are based upon the particular values as tabulated insofar as they relate to the total refrigeration required.

A study of the technical data in Tables 2 to 10 inclusive will show the floor to floor variation of typical office buildings. These values are totaled or averaged and compiled in the composite summary of Table 11 comparing the 10 air conditioning systems on an overall basis.

This composite summary reveals that insofar as population density is concerned this factor varies over a wide range from a minimum of 64 sq ft of floor area per person in Building B to 142 sq ft of floor area per person in Building A.

The average lighting in watts per square foot of floor area also varies over a wide range and is usually the amount of lighting required simultaneously with maximum sun effect on a given building. This particular value has a tendency to be increased due to possibility of widespread adoption of indirect lighting systems of extreme intensity which will have a definite bearing upon the required capacity of the air conditioning plant. The maximum average lighting load simultaneous with maximum sun effect is that tabulated for Building J or 2.97 w per square foot of floor area.

As for outside air supply furnished for ventilation purposes, the normal minimum infiltration will generally supply in a typical office building a liberal amount of outside air for maintaining an unusually high ventilation standard. Where the quantity of outside air is definitely controllable, as is also the population density within the space, the quantity of outside air introduced in cubic feet per minute per person can be more closely regulated. The windowless office Building I, in which infiltration can be reduced to an absolute minimum through the elimination of windows, was installed using 9.07 cfm of outside air per person against the maximum outside air quantity computed for Building F of 100 cfm per person.

The minimum outside air supply ranges from 0.44 changes per hour in the windowless Building I to 3.88 changes per hour in Building F where climatic conditions economically permitted the use of 100 per cent outside air.

The air circulated through the conditioned space ranged from 0.71 cfm per square foot in Building A to 1.66 cfm per square foot in Building H, the latter quantity being required by the unusually large portion of glass in the walls and the large exposure to sun effect.

The circulated air change ranges from a maximum circulation of 6.6 min air change for Building H to a 17.9 min air change for Building A.

The cubic volume per ton of calculated refrigeration load ranges from a minimum of 2880 cu ft per ton to a maximum of 6050 cu ft per ton in

TABLE 11—COMPOSITE SUM-

	BUILDING A	BUILDING B	BUILDING C	BUILDING D
Total Number of Floors	10	18	13	12
Number of Floors Conditioned	8	15	9	11
Outside Air Design Conditions	100 F DB–79 F WB	100 F DB–79 F WB	95 F DB–75 F WB	95 F DB–75 F WB
Inside Air Design Conditions	80 F DB–50% RH	80 F DB–50% RH	80 F DB–50% RH	80 F DB–50% RH
Floor Area, Sq Ft	159,712	80,960	45,000	97,700
Room Volume, Cu Ft	2,159,900	1,078,371	425,000	817,500
Number of People	1,195	1,266	650	1,320
Floor Area per Person, Sq Ft	142.0	64.0	69.2	74.0
Watts of Light	37,330	84,250	45,000	42,000
Avg Watts per Sq Ft of Floor Area	0.22	1.04	1.0	0.43
Min. Outside Air, Cfm	25,000	41,605	10,500	24,200
Min. Outside Air, Cfm per Person	20.9	32.9	16.2	18.3
Min. Outside Air, Changes per Hour	0.70	2.32	1.48	1.78
Internal Sensible Tons	264.0	173.1	59.4	98.4
Internal Latent Tons	77.3	16.1	8.3	16.8
Total Internal Tons	341.3	189.2	67.7	115.2
Total Tons ÷ Sens. Tons = Avg Int. Mult	1.29	1.09	1.14	1.17
Circulated Air, Cfm	120,000	110,179	38,000	82,300
Cfm per Sq Ft of Floor Area, Avg	0.71	1.36	0.85	0.84
Circulated Air Change, Minutes Avg	17.9	9.2	11.2	9.9
Tons Refrigeration Required	454.68	373.5	92.1	170.1
Volume per Ton, Cu Ft	4,750	2,880	4,620	4,800
Floor Area per Ton, Sq Ft	374	217	488	574
Equipment Summary				
Refrigeration—Compressors	2–14″x10″ 4 Cyl.	15″x13½″ 4 Cyl. 15″x13″ 2 Cyl.	6103 FE	2–12½″x10″ 2 Cyl.
Condensers	2–43½″x16′	4365 and 6065	3 EFB 2000	2–12″x15′
Coolers or Direct Exp.	2–43¼″x16′	3364 4864		38″x16′
Cooling Tower	Forced Draft	Forced Draft	None	Forced Draft
Air Conditioning Equipment	2 Base. Washers Supply Floors 2–8. Penth. Washer Supplies 9th Floor	Base. Washer Supplies Floors 1–7. Penth. Washer Supplies 8–1st. Penth	Washer on Roof Supplies Floors 4–12	2 Units on 2nd (1st-3rd) 2 Units on 5th (4th–6th) 2 Units on 8th (7th–9th) 2 Units on Roof (10th and 11th)
Zones	2 Recirc. Fans Each Floor 2–8 Separate Supply to 9th Floor	3 Recirc. Fans for Floors 1–10 4 Recirc. Fans for Floors 11–1st Penth	By-pass Control at Washer Supplementary Vol. Control Each Floor	By-pass Control at Each Unit Supplementary Vol. Control Each Floor
Heating	O. S. Air Temp. Indirect	O. S. Air Temp. Indirect	Direct Rad. O. S. Air Temp.	Direct Rad. O. S. Air Temp.
Automatic Controls	Pneu.	Pneu.	Diff.-Pneu.	Diff.-Elec.
Filters	Throw-Away	Throw-Away	Throw-Away	Cleanable
Miscellaneous				

* See under technical data by floors.

MARY OF TECHNICAL DATA

BUILDING E	BUILDING F	BUILDING G	BUILDING H	BUILDING I	BUILDING J
20	7	17	13	4	15
19	5	15	12	4	14
100 F DB–76 F WB	100 F DB–70 F WB	95 F DB–78 F WB	95 F DB–78 F WB	95 F DB–76 F WB	95 F DB–78 F WB
80 F DB–50% RH	80 F DB–45% RH	80 F DB–50% RH	80 F DB–45% RH	a	80 F DB–50% RH
83,248	54,100	60,910	106,897	168,128	190,510
933,795	616,740	662,540	1,169,105	2,492,106	2,071,500
755	400	635	1,515	2,015	2,084
110.2	135.2	95.8	70.5	83.7	91.3
96,800	6,000	110,000	223,041	406,944	565,040
1.16	0.11	1.81	2.09	2.42	2.97
17,800	40,000	15,509	29.040	18,245	53,020
23.6	100	24.4	19.2	9.07	25.4
1.14	3.88	1.41	1.49	0.44	1.54
141.0	72.2	94.0	242.8	305.0	295.4
24.7	5.1	19.0	25.5	31.5	29.7
165.7	77.3	113.0	268.3	336.5	325.1
1.18	1.07	1.20	1.11	1.10	1.10
93,200	40,000	62,522	177,400	239,600	149,350
1.12	0.74	1.03	1.66	1.42	0.78
10.0	15.4	10.6	6.6	10.4	13.8
217.4	144.3	169.8	366.4	411.7	515.6
4,290	4,260	3,900	3,180	6,050	4,020
383	375	359	291	408	369
7108–F	6751–F	14″x10″ 2 Cyl.	2–7108–F	2–14″x10″ 2 Cyl.	2–610F 2F–
798–F	...	9″x7½″ 2 Cyl.	...	1–15″x10″ 2 Cyl.	...
31,251	1,202	43″x16′	41,251	3–38″x16′	81,251
3,806	Direct Exp.	38″x16′	4,303	3–28″x16′	2–4306
Forced Draft	Atmospheric	None	Forced Draft	None	Forced Draft
Air Cond. Units on Each Floor	F-12 and Water Coils in Basement Supply Floors 2–6	4th Floor Washer Supplies Floors 1–5. 10th Floor Washer Supplies 6–15 Floors	C. S. Air Washer Supplies 1st Floor	Washers in Base. Supply Base. to 3rd Floors	3 Base. Washers Supply Base. 2nd Floor Washers on 14th and 15th Serve Floors 3–13
Air Cond. Units Each Floor Surface Type	2 Zones on Each Floor Volume Control	2 Recirc. Fans Each Floor	1 Recirc. Fan per Floor Supply Volume Control 4 Zones Each Floor	1 Washer Base. and 1st Floor 1 Washer 2nd, 3rd Floor C.S. Bypass Control	Air Mixing Units in Rooms Bypass Control and Supply Manual Control
Direct Rad. O. S. Air Temp.		Direct Rad. O. S. Air Temp.	Direct Rad. O. S. Air Temp.	O. S. Air Temp. Indirect	Direct O. S. Air Temp. Indirect
Pneu.	Pneu.	Diff-Pneu.	Pneu.	Diff-Pneu.	Diff-Pneu.
Throw Away		Throw Away	Throw Away	Throw Away	Auto. Oil
	C. T. Water Used For O. S. Air Precooling				

Buildings B and I respectively. Thus, the windowless office building, Building I, which would not have been possible without air conditioning, indicates the reduction in first cost for air conditioning equipment if the building factor is controlled. A reduction in operating cost per cubic foot of conditioned space would likewise result with an insulated building of the windowless type such as Building I. A variation in floor area per ton ranges from 217 sq ft per ton in Building B up to 574 sq ft per ton in Building D.

CONCLUSIONS

The analyses and summaries herein presented are descriptive of office building air conditioning installations in ten buildings located throughout the United States.

No two of these buildings were designed in exactly the same manner. It is possible by making use of the various alternate methods of installing an air conditioning system in an office building to make such an installation in many different ways, when it is considered that there are many types of equipment available with many alternates which can be offered for accomplishing a definite specified and guaranteed result.

The variables in application factors which are beyond control of the designing engineer are numerous and it would thus be futile to specify a given set of rigid standards to govern the installation of air conditioning systems in office buildings.

It does appear to be practical, however, to compare the over-all average composite data selected from a large group of representative large sized installations to establish the limits which have been successfully applied in experience to produce for the purchasing public a satisfactory result in air conditioning.

With the many types of air conditioning apparatus available, and the numerous application factors encountered, the air conditioning engineer will find the office building air conditioning system a test of ingenuity and resourcefulness if the most economical system is to be selected for each and every installation.

No. 1085

VENTILATING THE LINCOLN VEHICULAR TUNNEL

By CHARLES W. MURDOCK * (*NON-MEMBER*), NEW YORK, N. Y.

WITH the opening of the South Tube of the Lincoln Tunnel in December, 1937, a second great motor vehicle tunnel under the Hudson River was made available for the ever-increasing traffic between New Jersey and New York. Located in Manhattan's mid-town area the new tunnel is the third trans-Hudson River vehicular crossing operated by the Port Authority, the others being the Holland Tunnel, completed in 1927, and the George Washington Bridge, opened in 1932.

Ground was broken on May 17, 1934, for this crossing, which was known as the Mid-town Hudson Tunnel until officially named the Lincoln Tunnel in April, 1937. It will have two separate tubes when completed, each tube providing for two-lane traffic in one direction. Until the North Tube is completed, the South Tube is to carry two-way traffic and thereafter will be used for traffic East-bound from New Jersey while the North Tube will carry the West-bound traffic to New Jersey.

The strategic location of the Lincoln Tunnel should provide considerable relief for the serious traffic problems of New York City, since with the completion of the Queens Tunnel, now being built under the East River at 38th Street by the City of New York, and the proposed crosstown vehicular tunnel through the heart of the Mid-town section, a direct underground highway from New Jersey and the West and South to Long Island will be provided, with surface connections on both east and west sides of the City.

While in general design and construction details the new tunnel is similar to the Holland Tunnel, its larger tube diameter provides a wider roadway and air ducts of larger cross-section. The principal divergence in the general scheme of the ventilating systems of the two tunnels is that the Lincoln Tunnel has three ventilation buildings, two in New York and one in New Jersey, instead of four as were built for the Holland Tunnel, which has two ventilation buildings on each side of the river.

GENERAL PLAN OF TUNNEL

The Lincoln Tunnel passes under the Hudson River as shown in Fig. 1, from West 39th Street in Manhattan to the New Jersey shore where it bores

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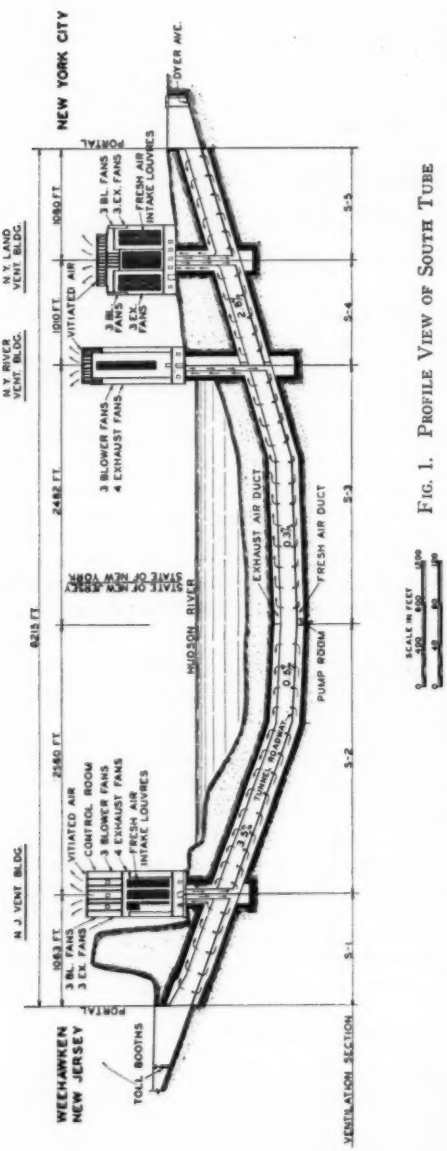

FIG. 1. PROFILE VIEW OF SOUTH TUBE

through Kings Bluff and then turns southward in a large radius curve, emerging on a plaza located between the Hudson County Boulevard East and Park Avenue in Weehawken. Tolls will be collected at toll booths located on this plaza. Connections with Union City, North Bergen and neighboring towns are provided. A new express highway which loops to the north and west from the plaza is being constructed and when completed will connect with Routes 1 and 3 of the N. J. Highway system.

The New York Plaza of the South Tube is located in the westerly half of the block bounded by Ninth and Tenth Avenues and West 38th to West 39th Streets. Access to the plaza is provided by Dyer Avenue, a new street between Ninth and Tenth Avenues extending from 34th Street to 42nd Street.

The under river tubes have an outside diameter of 31 ft, as indicated in Fig. 2, which is 1 ft 6 in. larger than the tubes of the Holland Tunnel and provides a roadway width between curbs of 21 ft 6 in. which is 1 ft 6 in. wider than the roadways in the older tunnel. Roadway headroom throughout is approximately 13 ft 7 in., which permits a safe operating clearance of 13 ft.

The South Tube is 8215 ft long between portals and consists of 240 ft of steel bent construction and 785 ft of steel lined circular rock tunnel on the New Jersey side, 5920 ft of cast iron lined shield driven tunnel under the

river and 450 ft of steel lined rock tunnel and 560 ft of steel bent tunnel to the New York Portal. The remaining 260 ft are contained within the limits of the ventilating shafts. The North Tunnel, now under construction, will have a total length between portals of about 7400 ft as its New York Portal will be located about one block nearer the river.

The maximum depth of the top of the tubes below mean high water is 75 ft and the roadway is 97 ft below the river at that point. The two tubes in the river section are to be 75 ft apart, center to center, and will have a minimum covering of 20 ft of silt. Among the more interesting constructional features of the tunnel might be mentioned the special lighting arrangements, the glass tile ceiling and the vitrified brick roadway surfacing, as shown in Fig. 3, the latter two items being new departures in vehicular tunnel design. The following tabulated data are of general interest:

Total Length (South Tube)	8,215 ft
Length of River Section Center to Center of Ventilation Buildings	5,062 ft
Number of Ventilation Buildings	3
Number of Fresh Air Fans—South Tube	15
Number of Exhaust Air Fans—South Tube	17
Maximum Fresh Air Supplied—South Tube	1,736,000 cfm
Maximum Operating Horsepower of Fan Motors—South Tube	2,237

DESCRIPTION OF THE VENTILATING SYSTEM

The ventilating system of the South Tube is essentially the same as that so successfully employed in the Holland Tunnel, and is generally known as the *Transverse Flow* type. This system has been used in The Detroit-Windsor Tunnel, the George A. Posey Tunnel in Oakland, Calif., the Boston Traffic Tubes, and the Antwerp Tunnel, and will be used in the Queens Tunnel now being constructed under the East River.

The fresh air from the blower fans flows through a duct below the roadway slab, which is illustrated in Fig. 4. Air is introduced into the roadway area through a continuous expansion chamber and a narrow adjustable slot located above each curb and connected to the duct below by flues 3½ ft wide and 6 in. high, spaced from 10 to 15 ft apart throughout the tunnel length. A uniform distribution of the fresh air throughout the tunnel length is obtained by adjusting the asbestos slide in each flue inlet according to the static pressure in the duct.

The exhaust duct is formed by the ceiling slab and the upper part of the tube shell, the vitiated air being withdrawn through adjustable cast iron ports, placed in pairs, one on each side of the centerline, and spaced at intervals of from 10 to 20 ft longitudinally. These exhaust port castings vary in size, and in addition have adjustable stainless steel slides so that the area of each port opening can be set as required by the static pressure in the exhaust duct at the point where the port is located.

The tunnel is divided into five ventilation sections, each section being served by one fresh air duct and one exhaust air duct which ascend vertically at the shafts to the blower and exhaust fans located in the ventilation buildings above. The exhaust air ducts lead to air-tight rooms in which the exhaust fans are placed, and the vitiated air is discharged to the outside atmosphere above the roof of the building.

The chief advantage of this method of ventilation is the practically complete elimination of longitudinal air currents. It is felt that a longitudinal flow of air in the space used by vehicles presents an undue hazard if the tunnel is one used by a large volume of traffic and the velocity of flow is such as would

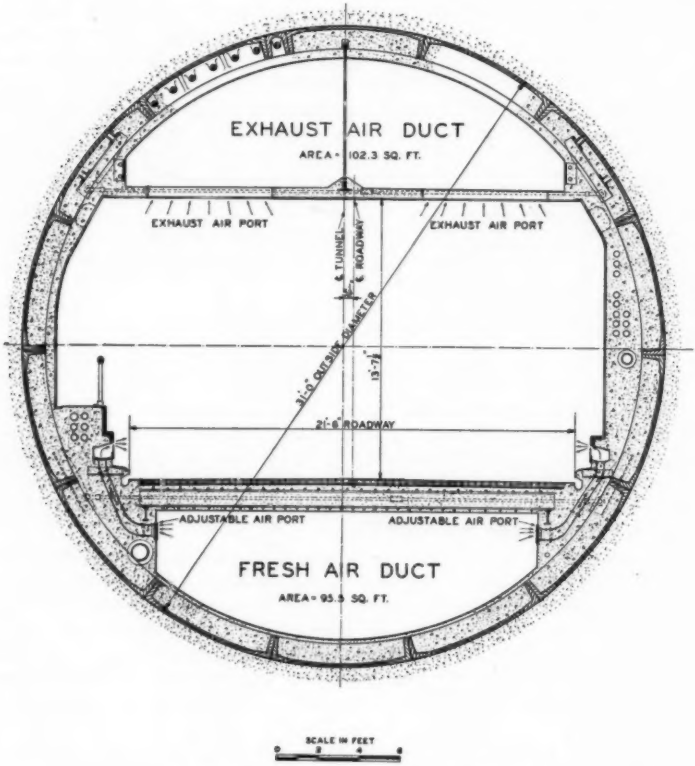

FIG. 2. CROSS-SECTION OF RIVER TUBE

rapidly spread smoke or other fumes. The chief disadvantages are the higher initial and operating costs.

Experience in the operation and maintenance of the Holland Tunnel has resulted in improvements in the ventilating equipment and its arrangement in the buildings, and has shown how careful attention to the many minor details simplifies the maintenance problem. In the design of the various parts of the duct systems particular attention has been directed to the improvement in details of construction and the reduction of resistance losses. All elbows have liberal radii and where conditions warrant it they are provided with splitting

vanes. Changes in area or shape are as gradual as possible, especially in expanding ducts. The areas of the exhaust ducts at the point where they enter the exhaust fan rooms are large in order that the expansion loss at those points will be small. Access to all tunnel ducts has been provided at the ventilating shafts, making it unnecessary to use the manholes in the roadway. The adjustable slide of the fresh air ports is accessible from the duct so that it is unnecessary to remove the expansion chamber face plate to make a change in the port area.

AIR REQUIREMENTS FOR VENTILATION

The determination of the amount of air required to ventilate the tunnel was based on the extensive Tunnel Gas Investigations conducted by the U. S. Bureau of Mines in 1920 and on similar tests made in 1934 at the Holland Tunnel which indicated that there has been comparatively little change in the quantity of carbon monoxide produced by motor vehicles since the earlier tests were conducted. The Bureau of Mines tests have been fully described in a bulletin[1] and summaries have been given in other literature. On the basis of these tests and actual experience in tunnel operation, the maximum rates of fresh air supply for the new tunnel were determined as follows:

TWO-WAY TRAFFIC

Section on Grades	200 cfm per linear foot of tunnel
Level Sections	225 cfm per linear foot of tunnel

ONE-WAY TRAFFIC

Sections on Up-grades	275 cfm per linear foot of tunnel
Section on Down-grades	150 cfm per linear foot of tunnel
Level Sections	225 cfm per linear foot of tunnel

These rates provide a margin of safety above the actual requirements for the safe ventilation of the tunnel at its maximum traffic capacity. However, in case of a fire or traffic jam this excess air supply will insure the rapid removal of the smoke and gases.

Since the South Tube will be used for two-way traffic until completion of the North Tube, when it will be used only for one-way traffic, all calculations used in the design had to be based upon the two different rates of air supply. When the South Tube is converted to one-way traffic operation, new settings of the fresh and exhaust air ports will be required. In addition the bulkhead between the two exhaust ducts under the river will be moved 300 ft to the east to properly equalize the performance of the fans connected to these ducts.

In all computations of air flow the density of the fresh air has been taken as 0.08 lb per cubic foot and for the exhaust air 0.0784 lb per cubic foot, these factors having been determined from a study of the U. S. Weather Bureau Records for a period covering eight years. The density of the exhaust air has been calculated by assuming a 10-deg rise in temperature of the air in the tunnel due to the heat in the motor exhaust gases.

AIR MEASUREMENT APPARATUS

To provide for the testing of the fans in actual operation and to check the velocity and static pressures in the ducts against the calculated values used in

[1] Ventilation of Vehicular Tunnels, by A. C. Fieldner, Yandell Henderson, J. W. Paul, R. R. Sayers, et al. (A.S.H.V.E. JOURNAL, January-December, 1926).

the design, permanent piping connections for the air measurement equipment have been incorporated in the walls of the fresh air and exhaust ducts. These measurement stations which are spaced at distances varying from 300 to 400 ft throughout the tunnel length, consist of piezometer rings with plates set in the sides of the ducts and a second pipe terminating at the duct in such a way that a Pitot tube can be located at the center of gravity of the duct area. The piping extends to a box in the wall above the sidewalk where connections to portable manometers can be made by means of rubber tubing. In addition the fresh air duct has a series of ferrules set in the roadway slab while the exhaust air duct has ferrules in the tunnel ceiling. Pitot tubes can be inserted

FIG. 3. INTERIOR VIEW OF TUNNEL ROADWAY

through these ferrules and readings taken at the proper points of the duct cross-section as is done in testing fans. From these readings the volume of air flowing in the ducts and the static pressures can be calculated and also a center constant established by means of which future determinations of the volume of air flowing in the ducts can be made without any interruption of traffic. This arrangement, in conjunction with the carbon monoxide recorders, makes it possible at any time and under any traffic condition to make a field check on the amount of carbon monoxide being liberated in the tunnel.

THE VENTILATION BUILDINGS

The location of the three ventilation buildings housing the fan equipment and providing the necessary air intakes and outlets for the system was determined by a study of the various factors involved. Comparative estimates of initial and operating costs indicated that two buildings for the entire system would be more economical than either three or four, provided that they could

be placed at fairly definite positions in relation to the tunnel length. The required position in New York was entirely impractical and in New Jersey the probability of construction difficulties made it seem advisable to locate that building further west where a rock foundation was available. These considerations made the three building layout the obvious solution.

The New York Land Ventilation Building is located at 39th Street and Eleventh Avenue and is built on foundations which extend below the tube and form the shaft to the tunnel. This building is approximately 135 ft by 52 ft in size and is four stories high with the exhaust air stack extending up another 30 ft above the roof making the total height of the building above

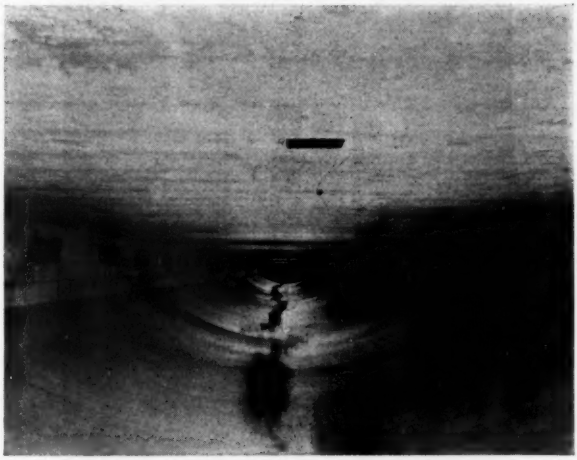

FIG. 4. SECTIONAL VIEW OF FRESH AIR DUCT

the street level 116 ft. This building, unlike the other ventilation buildings, was constructed as a complete unit to accommodate the fans for ventilating both the north and south tubes between the river shafts and the portals. The fans for this section of the North Tube have been installed and will remain inoperative until this tube is completed. For the South Tube there are six blower fans located on the second, third and fourth floors which draw the fresh air through bronze louvers set in the south and east walls of the building and force it into the fresh air ducts of the tube. The louvers extend almost the full height of the building from about the second floor level to just below the roof. The six exhaust fans which are located on the same floors draw the vitiated air from the tunnel ducts and discharge it to the atmosphere through the evasé stacks extending above the roof. The intake louvers are not equipped with motor operated rolling shutters as in the other two buildings since there is no appreciable fire hazard in the surrounding buildings and provision for closing off the intakes was not considered necessary. All fans in this building are chain driven and the motors are set on concrete bases at con-

venient heights above the floor making them accessible for maintenance and repairs.

The New York River Building is situated in the slip at the foot of West 39th Street. It is built on the concrete filled steel caisson of the river shaft, the east face being a few feet outside of the U. S. Bulkhead Line. The lower portion of the caisson was built in a shipyard, floated to the site and then sunk to bedrock. The building above, being larger in dimensions than the

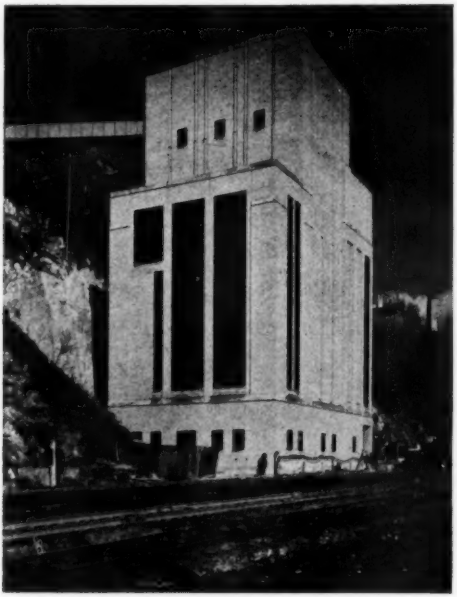

FIG. 5. EXTERIOR VIEW OF NEW JERSEY VENTILATION BUILDING

caisson foundation, is cantilevered out on all sides above the mean high water level of the river.

This building is about 62 ft 6 in. by 53 ft 4 in. in size and is six stories high with the top of the exhaust air discharge stack parapet 150 ft above the river. The intake louvers are located in the north, south and west walls of the building and extend from just above the third floor to near the roof. As a protection from the smoke of fires that may occur in the neighboring piers all the louvers have motor operated aluminum rolling shutters inside the building so arranged that they can be operated manually also. Each of the four exhaust and three blower fans is driven by two 2-speed motors. All fan motors are set on concrete bases above the main floors. The exhaust fan room floors were placed about 5 ft lower than the main corridor floor so that the direct connected motors driving these fans would be at a convenient working height. The New York River Ventilation Building was

built as a separate unit to house the fan equipment for the ventilation of the South Tube from this building to the Mid-river bulkhead. When the North Tube is completed there will be another ventilation building identical in size and shape located about 15 ft to the north. The two ventilation buildings will be connected by a cross passage at the tunnel level and also by an enclosed bridge at the sixth floor.

The New Jersey Ventilation Building, illustrated in Fig. 5 and diagrammatically shown in Fig. 6, is located about 1000 ft inshore from the U. S. Bulkhead Line and approximately 400 ft from the river's edge. It is 100 ft 6 in. by 63 ft 4 in. in size, 7 floors in height and extends to a total height of 150 ft above the ground level, the first floor being some 10 ft above mean high water of the river. It is situated close to the face of King's Bluff and is connected to the street on top of the bluff by a steel girder foot bridge at the seventh floor level. The louvers are built into the north, south and east walls of the building and extend from about the second floor to within 4 ft of the fifth floor. All the louvers are equipped with the same type of motor operated rolling shutters used in the New York River Building. There are six blower and seven exhaust fans in the building of which three of the blowers and three of the exhausters serve the land portion of the South Tube extending to the New Jersey portal. All the fans for the land section of the tube are chain driven. The three blower and four exhaust fans for the river section are direct driven by the same combination of two motors as in the New York River Building. The motors for all fans are set on concrete bases which are at accessible heights above the floor. The exhaust fans for the river section are in rooms having sunken floors, as in the New York River Building.

The New Jersey Ventilation Building, like the New York River Building, was built as a separate unit for the fan equipment of the South Tube. A new building will be built for the North Tube equipment on new foundations adjacent to the present building so that the two structures will appear as a single unit.

Dampers are provided in each duct between the fan and the common duct leading to the tunnel. These dampers are raised and lowered by means of chains running on sheaves operated by motors with worm gear speed reducers. The control of the damper motor is arranged so that when a fan is started the damper opens and remains open as long as the fan is running. When the fan is stopped the damper closes.

All the exhaust fans are provided with drains in the bottom of their housings to remove the rain water which may enter the discharge stacks.

FANS AND MOTORS

The total number of fans required for the operation of the South Tube is 32 of which 15 are blowers and the other 17 exhausters. One of the fans connected to each duct section is a spare available for use in case of breakdown and is not required to be operated for the maximum ventilation of the tunnel. The fans are all of the double width, double inlet, centrifugal type with backwardly curved blades. The blowers are downward blast discharge and the exhausters upward blast. The housings are of standard steel plate construction and were fabricated in sections to facilitate installation and maintenance. Fan

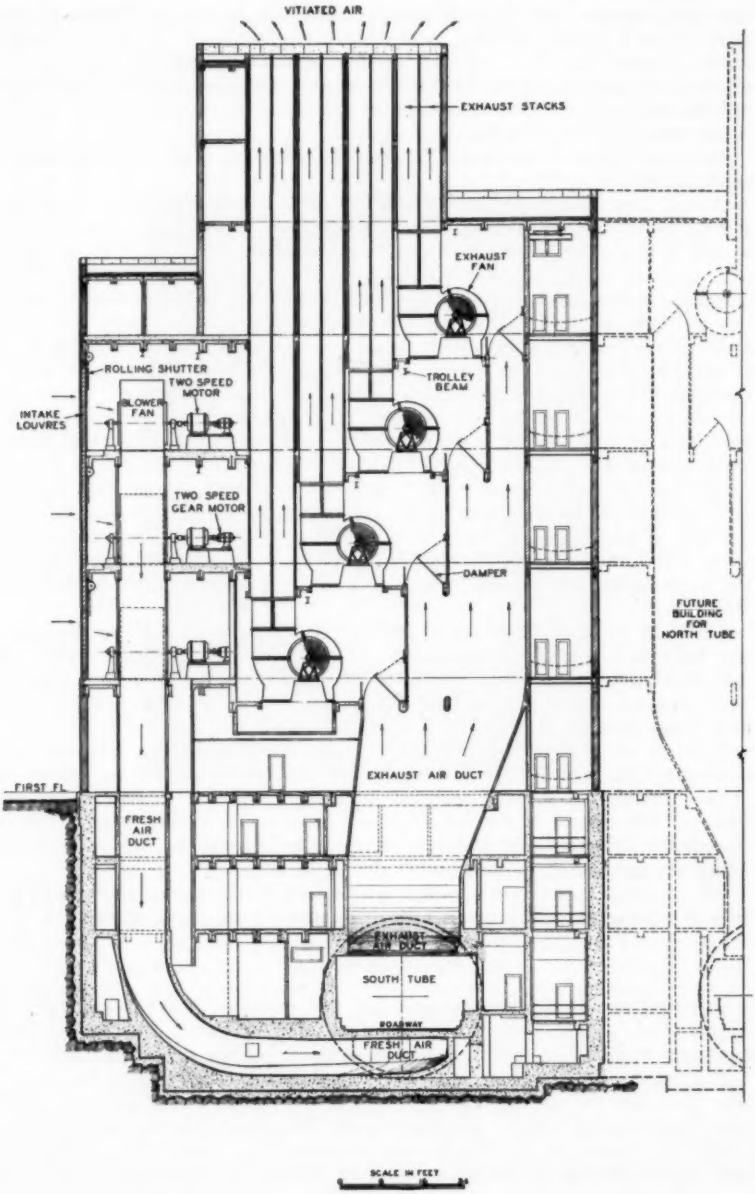

FIG. 6. SECTION THROUGH NEW JERSEY VENTILATION BUILDING

inlets are equipped with vaned inlet cones and electrically welded rigid wire guards. The fans are of standard construction with heavy cast-steel cone hubs and the wheels have 24 blades on each side of the center plate. Shafts of extra stiffness were required by the specification so that the maximum deflection between the bearings would not exceed 0.005 of an inch per foot of length between bearings. Fan bearings are of the roller type, one bearing being fixed and one full-floating for each fan.

Table 1 shows the number of blower and exhaust fans required for the South Tube together with the section of the tunnel duct to which they are connected, the building in which they are located, the specified volume of air and static pressure when operating at high speed.

Inasmuch as the fans, both blower and exhaust, will be used in various combinations they are of such design that they will operate successfully in parallel. The specifications further called for fans which under any condition of pressure or volume would not cause overloading of the motors and

TABLE 1—BLOWER AND EXHAUST FANS FOR SOUTH TUBE

No. of Fans	System	Duct Section	Location Ventilation Building	Volume Cfm per Fan	Static Pressure In. of Water
3	Blowers	S B 1	New Jersey	106,000	0.500
3	Exhaust	S E 1	New Jersey	108,000	0.375
3	Blower	S B 2	New Jersey	272,000	3.000
4	Exhaust	S E 2	New Jersey	185,000	4.000
3	Blower	S B 3	New York River	280,000	3.750
4	Exhaust	S E 3	New York River	185,000	4.000
3	Blower	S B 4	New York Land	105,000	0.750
3	Exhaust	S E 4	New York Land	108,000	0.375
3	Blower	S B 5	New York Land	105,000	0.750
3	Exhaust	S E 5	New York Land	108,000	1.000

which with a small increase in static pressure would not result in a large decrease in volume of air delivered.

The performance characteristics of the fans are such that high efficiency can be maintained over a wide range of air volume delivery. The maximum horsepower occurs at a point near the point of maximum efficiency and the fans in each case were selected to operate near this point.

In order that fans, motors and transmissions would be of the highest grade and would prove efficient and economical, not only in first costs but also in operating costs, the contract for this equipment required bidders to specify the *Lump Sum Cost* and the *Guaranteed Cost of Operation*, the contract being awarded to the bidder whose sum of these two costs was lowest. The Guaranteed Cost of Operation was determined by the following formula:

$$\text{Cost of Operation (in dollars)} = 0.22 \times H \times 1500$$

in which H is the combined power consumption in kilowatts of all fans, except the spares, when operating at high speed, the factor 1500 is the probable cost, in dollars reduced to present value, of 1 kw used continuously during the probable life of the equipment. The factor 0.22 is the ratio of the anticipated annual power consumption to the power consumption of all fans, except the

spares, if continuously operated at the high speed. In arriving at the value for *H* the efficiencies of the drive transmission were taken as 95 per cent for chain drives and 100 per cent for those direct connected. This was done because it was felt that there were not available at that time sufficient data on the efficiency of chain drives to justify the inclusion of that item. Specifying the efficiencies to be used removed this uncertain factor from consideration in comparing bids.

The records of the Holland Tunnel show that the fans, all of which are driven by wound rotor motors, are operated at approximately one-half speed for about 80 per cent of the time and that satisfactory ventilation could be maintained for a considerable part of this time with the fans operating at lower

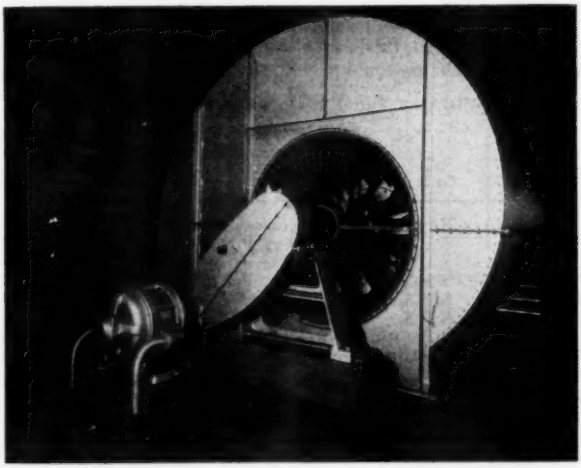

FIG. 7. VIEW OF BLOWER FAN WITH CHAIN DRIVE

speeds. However, the inherent instability and low efficiency of that type of motor at extremely low speeds made a further reduction impractical. Therefore, in selecting the equipment for the Lincoln Tunnel squirrel cage motors were specified since it was believed that the proper operating speeds of the fans could be determined in advance of actual operation of the tunnel and that by selecting definite motor speeds and types of drive a high efficiency could be maintained at all speeds.

The number of fan speeds and the type of transmission best suited for each duct section were determined by an analysis of the expected traffic load. As the traffic load of the tunnel will be subject to daily and seasonal fluctuations, a large range in speed variation is necessary in order that the volume of air supplied and exhausted can be regulated as the traffic load varies. When the air requirements for each duct section were plotted on a graph against the traffic load for a yearly cycle of operation and evaluated to kilowatt input, it was found that for the duct section in the land portions of the tunnel, two-

speed motors operating at full and one-half speeds would be the most economical in combined initial and operating costs. For the fans in the river sections where the horsepower requirements are considerably higher due to the increased duct lengths, four fan speeds gave the optimum condition as to costs.

The motors for all fans on the land duct sections operate at synchronous speeds of 900 rpm and 450 rpm and drive the fans by means of chain drives enclosed in oil tight cases as illustrated in Fig. 7.

The direct-connected motors driving the fans for the river section ducts shown in Fig. 8 are a combination of two separate motors on one end of the fan shaft. In this arrangement a large two-speed motor is coupled through a flexible coupling to the fan shaft and a smaller geared two-speed

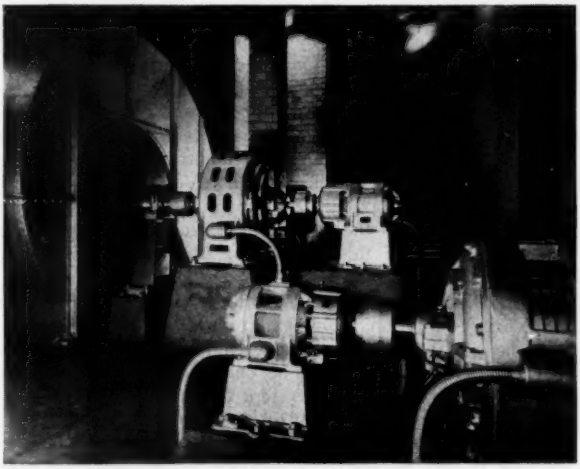

FIG. 8. VIEW OF BLOWER FAN WITH DIRECT-CONNECTED MOTORS

motor connected to the extended shaft of the larger motor through a flexible over-running clutch. The large motors run at synchronous speeds of 450 rpm and 300 rpm while the smaller geared unit has output speeds of 225 rpm and 112 rpm. When the larger motor is operating the fan the geared motor is cut out of service by the over-running clutch and when the lower speeds are required the geared motor drives the large motor and the fan. With this drive arrangement low speeds are obtained at high efficiencies and without the use of expensive four-speed motors.

The geared motors are of the single reduction planetary type. All motors for the fans operate on 440 volt, 3 phase, 60 cycle current.

All the fan equipment was painted with a special aluminum paint selected to provide the maximum resistance to the corrosive effect of the salty atmosphere prevalent on the New York water front.

CONTROL SYSTEM

The electric power for the operation of the tunnel is purchased from public

utility companies on both sides of the river, the current being supplied over 3 independent feeders from each company, each of the 6 cables having sufficient capacity to handle the entire tunnel load, thereby providing adequate insurance against power failure. The three 13,200 volt New York feeders are cross-connected and two cables from this source of supply run through the tube to the New Jersey side. The feeders on that side, which are 4160 volt, are also cross-connected and two cables run from that source through the tube to the New York Ventilation Buildings. All the transformers and equipment of the New Jersey power service are suitable for the higher voltage so that if at some time in the future the voltage should be increased to that in New York it will only be necessary to change some of the connections.

A central supervisory control station is located on the sixth floor of the New Jersey Ventilation Building. The control board located in this room contains the switching equipment by means of which the operator has complete control over the entire mechanical and electrical equipment of the tunnel. The operator selects the source of power and distributes it to the various units as is desired, operates the fans at the desired speeds and controls the dampers, signal apparatus, and lighting system. Through the telephone system and the code call equipment the attendant can communicate with all the employees in the tunnel or the buildings. Indicators on the control board show the operation of all fans, pumps and auxiliary equipment and the position of the dampers. Potentiometers connected to the carbon monoxide analyzers show and record at all times the amount of this gas in the atmosphere of each of the five ventilation sections of the tunnel.

In addition to the central control station there is a local control board in each of the New York Ventilation Buildings, from which the fans in each of these buildings can be operated. Adjacent to each fan there is a local control panel from which that particular fan can be operated. The three controls are so interlocked that it is impossible to operate a fan from two points simultaneously.

An interesting feature of the control system is the manner in which the speed of the fan motors is controlled. As all motors are designed for full voltage starting it was necessary to provide some means of preventing the high counter torque which would result if a low speed connection were to be made while the fan and motor were running at a higher speed. The control consists of a small magneto which is gear driven from the end of the fan shaft. As the frequency of the current generated by the magneto varies directly with the speed the circuit can be made to produce a sharp increase in current at critical frequencies by proper adjustment of the capacitance in relation to reactance. Relays responsive to the increase in current cause the motor starting contactors to operate at the proper frequencies which correspond to the selected speed of the motor. The same magnetos also operate the calibrated volt meters which provide the speed indication on the supervisory switchboard.

HEATING

The heating of the buildings used in connection with the operation of the Lincoln Tunnel presented no serious problems. The two New York Ventilation Buildings, each of which has only one room that requires heating, have been equipped with electric heaters. The New Jersey Ventilation Building, which contains the supervisory control room, telephone room and toilets for the

employees, is heated by a two-pipe steam system and an oil-fired cast iron boiler with full automatic controls. Heat is also provided in a stairway in this building to protect the standpipe from freezing. All radiators are of cast iron, those in the control room being of the concealed convector type. The radiators are equipped with valves of the adjustable orifice type and condensation is returned to the boiler by gravity.

At the New York Plaza near the tunnel portal is the Emergency Garage, a two-story building having a garage on the first floor and offices above. This building has a heating plant which consists of a two-pipe steam system and an oil-fired cast-iron boiler. In the garage two ceiling type unit heaters have been installed. The offices have wall type legless cast-iron radiators. The roof of this building has a 1-in. thickness of cork over the concrete roof slab for insulation and the four glass skylights have been equipped with heating coils. The control of this system is also fully automatic, the operation of the oil burner being regulated by a thermostat.

The general operating offices of the Tunnel are situated in the Field Office Building adjacent to the New Jersey plaza and provide excellent facilities for the superintendent, the toll collectors, and the Tunnel police. The first floor of this building contains a large garage for the automotive equipment, wrecking cars and fire apparatus, as well as a machine shop for their maintenance. On the second floor are located the general offices, police headquarters, doctors' office, locker rooms, storerooms, toilets, washrooms and other facilities. The building is heated by a two-pipe steam system, with a vacuum return pump and a fully automatic oil-fired cast-iron boiler. All radiators in the offices and machine shop are of cast-iron, the legless type being used wherever possible. Piping has been run as direct as possible. Float trap drips were used at several points because clearances in the garage necessitated running considerable piping pitched against the steam flow. The garage on the first floor is heated by unit heaters placed near the ceiling. The entire roof is insulated with 1 in. of cork placed over the roof slab.

This building is located on the east side of the plaza directly opposite the line of toll booths which extend across the plaza. In planning the heating system for the building provision was made for heating the toll booths from the same system. An underground pipe gallery extends from the boiler room across the plaza. In the gallery are located the steam and return mains from which connections are provided to the unit heaters in the 14 toll booths.

Because of the various purposes for which the space in the Field Office is used the heating system was zoned to meet the variable heating demands. Four heating zones are used, the toll booths and the garage unit heaters each are on a separate zone and the second floor offices are divided into two zones. Each zone has its own thermostat which operates the oil burner controls and a motorized steam valve on its main near the boiler. Also, the motors of the unit heaters in the garage are operated by the thermostat in that zone. The control system is so arranged that if heat is required in any one of the zones the burner will start and the valve will open to admit steam to that zone without affecting the others.

The heating of toll booths by convection heaters has been unsatisfactory in the past because of the fact that the window being open for a large part of the time, there has been a cold stratum of air at the bottom of the booth while the upper part has been relatively overheated. As a result the toll collec-

tors have suffered from cold legs and feet. To overcome this objection it was decided to use a small unit heater with directional louvers in each booth. These heaters have a two-speed manual control. They are located under a shelf at one end of the booth, the center of the heater being 15 in. above the floor.

The booths which are only 6 ft 9 in. long, 2 ft 9 in. wide and 7 ft 4 in. high have glass windows on all four sides. A 1-in. thickness of cork insulation covered by a layer of rubber tile has been provided on the floor as a further aid to the comfort of the collectors. The roof is insulated with rock wool.

DISCUSSION

DR. A. C. WILLARD (WRITTEN): This paper is, in my opinion, one of the best and most complete descriptions of the latest practice in vehicular tunnel ventilation ever written. While the paper is purely descriptive, and makes no attempt to review or analyze any of the research and engineering data on which the design of this unique ventilation system is based, it is a most valuable contribution to the professional literature in this field.

I know of no major ventilation undertaking which received more exhaustive advance study, research and investigation than the ventilation of the Holland Tunnel which has been in successful operation since 1927. The new Lincoln Tunnel design is, as Mr. Murdock states, based directly on the earlier design with which he was intimately connected, and the experience resulting from the operation of the latter project.

It seems remarkable that no essential changes have been found necessary as a result of that experience in a period of 10 years of operation. Most remarkable is the fact that practically the same air quantities per automobile were found to be necessary in the new tunnel, as shown by recent corroborative tests of present day motor vehicles, as were found necessary in the old tunnel. This means that the average air contamination of our highways and city streets, due to carbon monoxide resulting from imperfect combustion in our automobile engines, has not changed essentially, by and large, in 10 years, at least, in the judgment of the tunnel ventilation engineers.

And finally, I wish to pay a tribute to the genius, the ability and the vision of Clifford M. Holland, the first chief engineer of the great Holland Tunnel, who through his high regard for the basic importance of research in dealing with novel engineering problems established the fundamental principles as well as the successful practice involved in our modern vehicular tunnel ventilation.

PROF. JAMES HOLT: I would like to ask Mr. Murdock one question. I have not seen the tunnel and I am going purely by Fig. 6, as shown. It shows exhaust fans with the exhaust duct and damper, the damper being located to the right of the fan, so that as the exhaust air rises into the exhaust fan room the natural flow of air would be spirally in a counter clockwise direction, whereas the intake of the fan shows that the air would then have to spiral in a clockwise direction. It would seem to me there would be considerable loss of kinetic energy by turning first in a counter clockwise direction and then in a clockwise direction at the inlet of the fan.

I am also wondering if any consideration was given to the aerodynamic treatment of the fan intake rooms to conserve as much of the kinetic energy as possible and avoid turbulence losses.

MR. MURDOCK: There is no question but that as the air comes into these fan rooms there is an additional loss due to the air distribution in the room itself. I am not prepared to say whether that is due to a whirling in the room or the general distribution of the air in the room, but nevertheless the loss exists. We know that and that is the reason why in our specifications we call for the testing of an exhaust fan as well as a blower fan, when we ask bidders to give a guaranteed cost of operation.

No. 1086

DRAFT TEMPERATURES AND VELOCITIES IN RELATION TO SKIN TEMPERATURE AND FEELING OF WARMTH

By F. C. HOUGHTEN * (*MEMBER*), CARL GUTBERLET ** AND EDWARD WITKOWSKI *** (*NON-MEMBERS*), PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines

DRAFTS are probably the source of more complaints directed against ventilating and air conditioning systems than any other defects. Notwithstanding this fact, the engineer has no way of evaluating what constitutes a draft other than his own personal feelings. There is even a lack of understanding of just what is meant by a draft. Literally the term means a movement of air. Probably most complaints of draft result from excessive movement of air.

The sensation commonly called a draft is a local sense of coolness affecting some part of the body, compared with the general feeling of warmth of the body as a whole. It is doubtful if a person can distinguish between a local sense of coolness caused by air motion and a local sense of coolness resulting from contact with air having a lower temperature. A person may also experience a local sense of coolness due to radiation from a portion of his body to a cooler surface, such as a cold wall or window. This feeling may also be interpreted as a draft.

From these considerations, the term *draft* as applied to conditions of the atmosphere within an occupied space may be accepted as meaning any local sense of cooling of a portion of the body,—caused either by an excessive movement of air of normal temperature, by air having a normal velocity but a lower temperature, by excessive radiation to a cold surface, or any combination of these three effects. The problem of determining what combinations of temperature and movement of air constitute drafts was recently assigned to the Research Laboratory by the A.S.H.V.E. Technical Advisory Committee OH-22 on Comfort Air Conditioning, consisting of the following: C. Tasker, *Chairman,* A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington R. E. Keyes, A. B. Newton and C. P. Yaglou.

* Director, A.S.H.V.E. Research Laboratory.
** Research Asst., A.S.H.V.E. Research Laboratory.
*** University of Pittsburgh Senior and Student Asst., A.S.H.V.E. Research Laboratory.

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1938, by F. C. Houghten.

The study carried out during the past year has been directed toward the development of an understanding of the effect of low temperature and velocities of air as they determined a sense of coolness or a draft on the ankle and the back of the neck. In order to carry out the study, a person seated in a room of uniform temperature and humidity and normal air velocities ranging from 15 to 25 fpm, as measured by the Kata-thermometer had air streams of various temperatures and velocities directed against either the back of the neck or the ankle. Observations were made of the temperature of the skin of the affected part and the sensation of coolness or comfort observed by the subject.

Figs. 1, 2 and 3 show the test arrangements for making these observations. Air from a source of which the temperature and moisture content was inde-

FIG. 1. SET-UP FOR DETERMINING ANKLE TEMPERATURE AND FEELING OF DRAFT

pendently controlled was supplied through a duct to the psychrometric chamber in which the subject was located. A Kata-thermometer located near the outlet, or nozzle, of the duct indicated the velocity of the air at this point and a thermocouple located near the same point indicated the air temperature. In carrying out studies of the neck the air velocity was adjusted to the desired value, indicated by the Kata-thermometer, without the subject being present and the orifice flow meter reading was noted. This nozzle air delivery was kept constant throughout the test. Also the temperature of the air at the outlet of the nozzle was maintained constant. After the skin temperature of the neck had been observed over a period of from 10 to 20 min, with little or no air flowing through the nozzle and with 70 F indicated by the thermocouple, the desired temperature and velocity were applied. The drop in skin temperature and the subject's expressed feeling of warmth of the affected part and of his entire body were recorded. In the study of the ankle each foot was kept in a corrugated paper packing box. Prior to the test, air was directed through the box so that the desired velocity was indicated by a Kata-thermom-

eter, located where the ankle was later to be placed, and the flow meter reading was noted. After the temperature of each ankle had been observed for 10 to 20 min, or until equilibrium was established in air of practically zero velocity and 70 F temperature, air of the desired temperature and velocity was directed against one ankle while the other ankle remained in the controlled condition. The drop in skin temperature of the affected ankle below that of the ankle in the controlled condition, during a 30 min exposure, was taken as a measure of the cooling effect of the draft.

This paper reports the results of 141 determinations of the cooling effect of drafts on the neck and ankle. Since any method of making skin tempera-

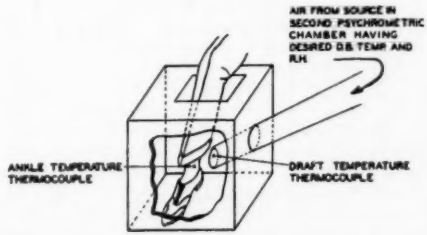
AIR FROM SOURCE IN SECOND PSYCHROMETRIC CHAMBER HAVING DESIRED D.B. TEMP AND R.H.
ANKLE TEMPERATURE THERMOCOUPLE
DRAFT TEMPERATURE THERMOCOUPLE

FIG. 2. TEST ARRANGEMENT FOR DETERMINING THE EFFECTS OF DRAFTS ON THE ANKLE

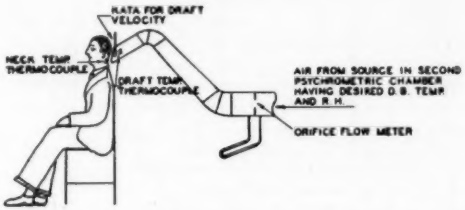
KATA FOR DRAFT VELOCITY
NECK TEMP THERMOCOUPLE
DRAFT TEMP THERMOCOUPLE
AIR FROM SOURCE IN SECOND PSYCHROMETRIC CHAMBER HAVING DESIRED D.B. TEMP AND R.H.
ORIFICE FLOW METER

FIG. 3. TEST ARRANGEMENT FOR DETERMINING THE EFFECT OF DRAFTS ON THE NECK

ture observations must necessarily have an important bearing on the interpretation of the results, a study of different methods is discussed. This investigation, and others previously made at the Research Laboratory, afforded an opportunity to make many observations of normal skin temperatures of several parts of the body which should be of general interest. The results of these observations are also reported in this paper.

MEASUREMENT OF SKIN TEMPERATURES

The measurement of the temperature of any surface is a difficult problem toward which a great deal of speculation and experimentation has been directed.

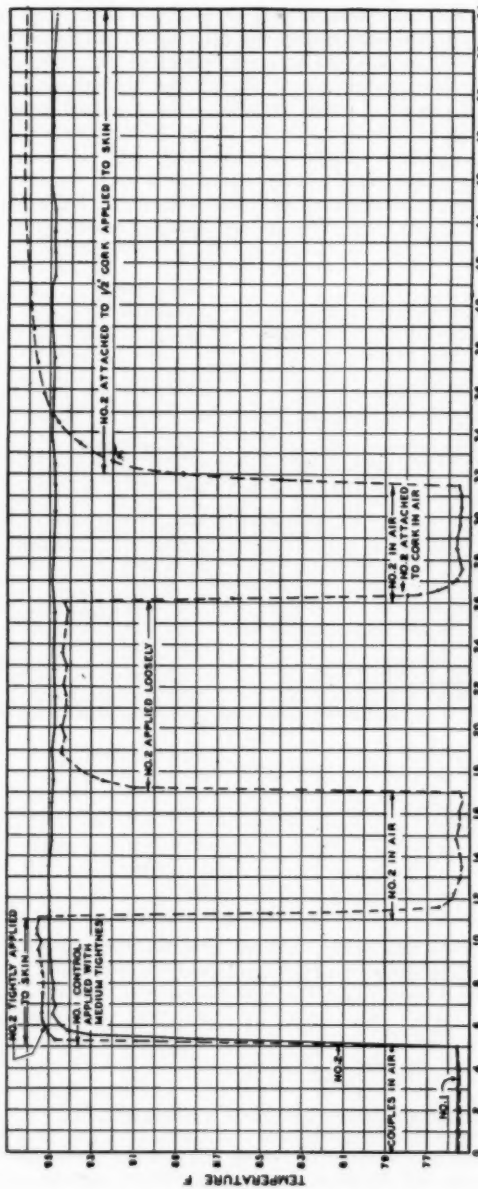

FIG. 4. RELATION BETWEEN METHOD OF APPLYING THERMOCOUPLE JUNCTION TO THE SKIN OF THE BACK OF THE NECK AND THE ELAPSED TIME TO REACH EQUILIBRIUM, THE OBSERVED SKIN TEMPERATURE AND THE CONSTANCY OF THE OBSERVED READING

All of the attendant difficulties and additional ones are met with in any attempt to determine the true temperature of the skin of the human body.

It is well to keep in mind that all that is done when measuring the temperature of any surface is to observe the temperature of the sensitive elements of a thermometer and pyrometer. Only through knowledge gained by experience can the sensitive elements of the instrument be so placed that it may be assumed to take on the temperature of the surface to be measured. In the case of observing the temperature of any surface, care must be taken, therefore, to place the sensitive thermocouple junction in contact with the surface so that, first, it will not interfere with the normal heat exchange between the surface and the air nor change the temperature of the surface; and, second, so that the instrument will take on the same temperature as the surface. Since both the surface to be measured and the couple junction are in contact with

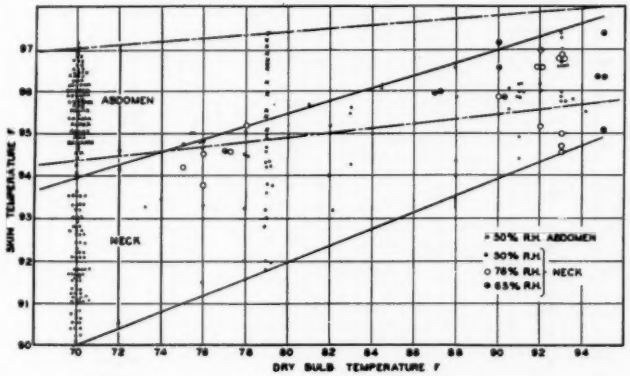

FIG. 5. RELATION BETWEEN DRY-BULB TEMPERATURE AND THE OBSERVED SKIN TEMPERATURE OF THE NECK AND ABDOMEN

the air, the junction will be jointly affected, under any circumstances, by the temperature of the surface and the air. The closeness and area of the contact between the junction of the thermocouple and the surface to be measured is, therefore, important. When the surface of which the temperature is to be measured is that of the human body, two additional factors affect the results: first, voluntary and involuntary movements of the body may affect the contact; and, second, the body surface dissipates heat by evaporation as well as by radiation and conduction. This evaporation must not be interfered with by the application of the measuring instrument.

Different methods of applying thermocouples to the body surface have been suggested and used by different investigators. Among these, the most common are the direct application of the thermocouple to the skin without covering. For curved sections of the body, this may be accomplished by drawing the junction firmly against the surface by the leads. For other locations, the junction and a portion of the leads on either side may be held in contact with

the surface by some form of web. In either case the temperature indicated will vary with the firmness of the contact—that is, whether the junction and the leads barely touch the surface, or whether they are pressed against the surface so as to be partly imbedded below the normal level of the surface.

Another method of applying the thermocouple is to attach it to some form of pad varying from ¼-in. to 1-in. in diameter. A cork pad suggests itself as the most advantageous type since it has low heat capacity and high resistance to heat flow. This cork pad application has definite advantages over the application of the bare couple. Any thermocouple, particularly when made of

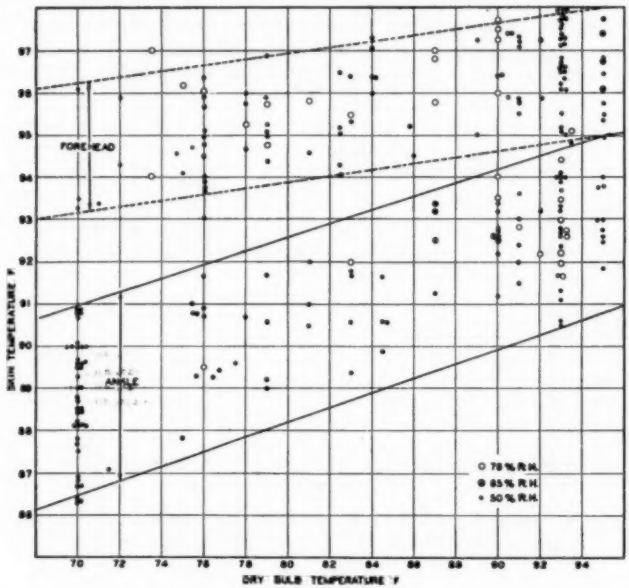

FIG. 6. RELATION BETWEEN DRY-BULB TEMPERATURE AND THE OBSERVED SKIN TEMPERATURE OF THE ANKLE AND THE FOREHEAD

fine wire, is subject to breakage near the junction. Applying the couple attached as an integral part of a cork or other pad greatly reduces the probability of breakage. The instrument, including the cork and the couple, can be applied to any part of the skin and held in place with uniform pressure with much less difficulty. Also this application, for reasons discussed, gives more constant and consistent results.

One important factor in observing skin temperature is the lag or lapse of time between the application of the instrument and the final equilibrium temperature observed. This depends upon the heat capacity and resistance to heat flow of the entire assembly of the junction and anything else in contact with it. A bare couple, made of fine wire, attached directly to the surface should, obviously, give the least lag.

It will be readily seen that the application of the bare thermocouple to the skin offers a minimum of interference to the normal heat loss from the skin at the point where the measurement is to be made, while the application of a pad offers greater interference. If cork were applied to any considerable area of the body surface, the resultant skin temperature would approach blood, or the normal body temperature of 98.6 F. The closeness of this approach would depend upon the area covered and the thickness of the insulation used. An attempt is made to overcome this effect by limiting the area of the skin covered. Obviously, the smaller the area covered, the less the effect. Only

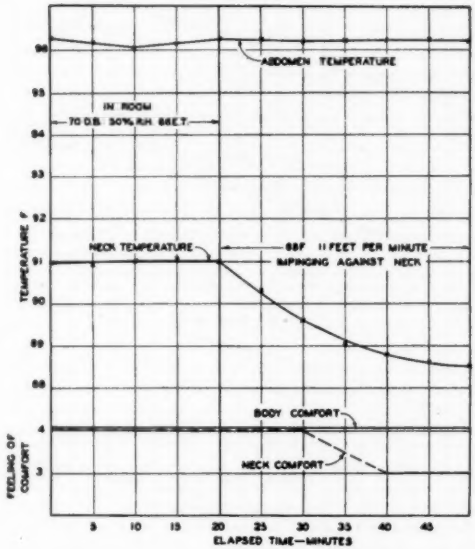

FIG. 7. RELATION BETWEEN ELAPSED TIME OF EXPOSURE OF NECK TO A DRAFT OF 68 F DRY-BULB TEMPERATURE AND AIR VELOCITY OF 11 FPM AND THE RESULTING SKIN TEMPERATURE AND FEELING OF COMFORT OF THE EXPOSED PART

experimentation can determine the magnitude of the resulting error accompanying the use of an instrument of this type.

In order to shed light on the relationship between the two methods of application and variation in the tightness of the application of the bare couple, a few observations were made which are plotted in Fig. 4. Thermocouple No. 1, after being in equilibrium with the air at about 76 F for a period of 5 min, was quickly applied to the skin with the usual degree of tightness in all such observations made at the Research Laboratory. A little more than half of the diameter of the No. 36 wire and junction was pressed

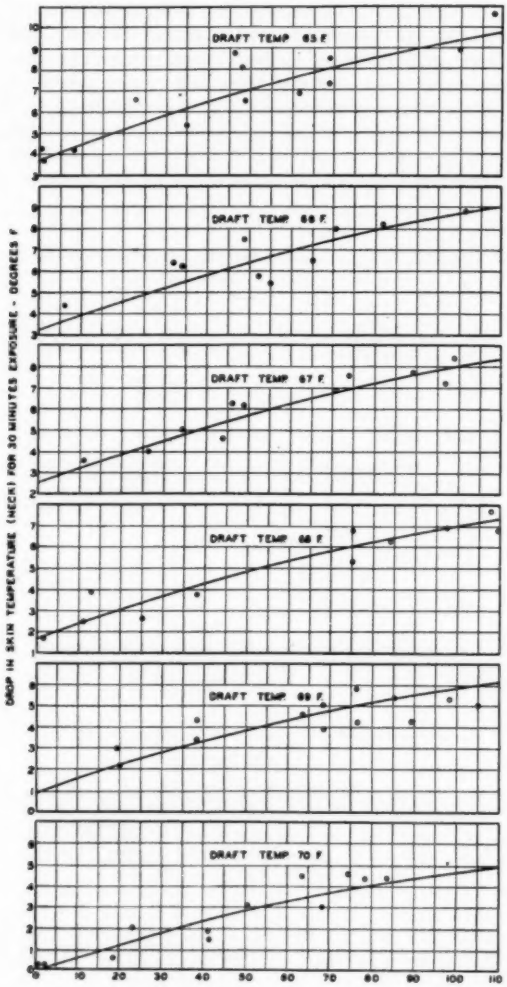
DRAFT TEMP 65 F
DRAFT TEMP 66 F
DRAFT TEMP 67 F
DRAFT TEMP 68 F
DRAFT TEMP 69 F
DRAFT TEMP 70 F
DROP IN SKIN TEMPERATURE (NECK) FOR 30 MINUTES EXPOSURE - DEGREES F
0 10 20 30 40 50 60 70 80 90 100 110

FIG. 8. RELATION BETWEEN AIR VELOCITY (DRAFT) AND DROP IN NORMAL SKIN TEMPERATURE OF THE NECK WITH A 30 MIN EXPOSURE FOR VARIOUS DRAFT TEMPERATURES. PERSONS SEATED AT REST IN A ROOM AT 70 DRY-BULB AND 50 PER CENT RELATIVE HUMIDITY (66 ET)

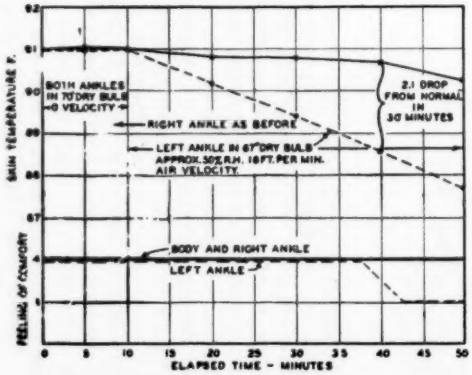
SKIN TEMPERATURE F.
FEELING OF COMFORT
BOTH ANKLES IN 70°DRY BULB =0 VELOCITY=
RIGHT ANKLE AS BEFORE
LEFT ANKLE IN 67°DRY BULB APPROX.50%R.H. 16FT.PER MIN. AIR VELOCITY
2.1 DROP FROM NORMAL IN 30 MINUTES
BODY AND RIGHT ANKLE
LEFT ANKLE
ELAPSED TIME - MINUTES

FIG. 9. RELATION BETWEEN ELAPSED TIME OF EXPOSURE OF THE ANKLE TO A DRAFT OF 67 F DRY-BULB TEMPERATURE AND AIR VELOCITY OF 16 FPM, AND THE RESULTING SKIN TEMPERATURE AND FEELING OF COMFORT OF THE EXPOSED ANKLE

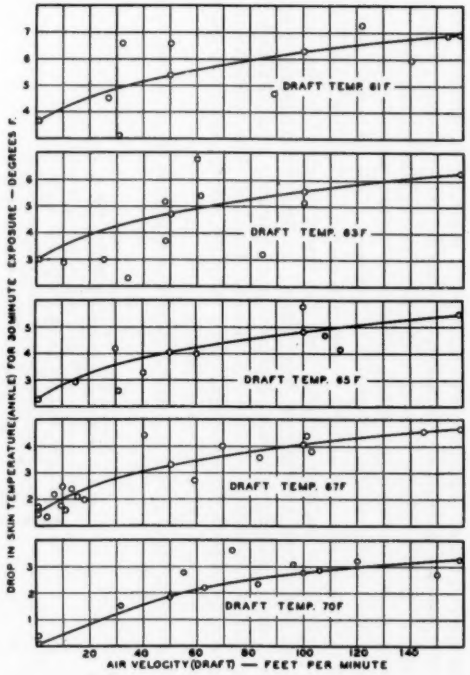
DROP IN SKIN TEMPERATURE(ANKLE) FOR 30 MINUTE EXPOSURE — DEGREES F.
DRAFT TEMP. 61F
DRAFT TEMP. 63F
DRAFT TEMP. 65F
DRAFT TEMP. 67F
DRAFT TEMP. 70F
AIR VELOCITY(DRAFT) — FEET PER MINUTE

FIG. 10 (*right*). RELATION BETWEEN AIR VELOCITY (DRAFT) AND DROP IN NORMAL SKIN TEMPERATURE OF THE ANKLE WITH A 30 MIN EXPOSURE FOR VARIOUS DRAFT TEMPERATURES. PERSONS SEATED AT REST IN A ROOM AT 70 F DRY-BULB AND 50 PER CENT RELATIVE HUMIDITY (66 ET)

below the normal level of the skin surface. It will be noted that this application gave complete equilibrium in about 1½ min and a constant indicated skin temperature thereafter. It should be observed that in these applications, the subject refrained more than normally from movements of his body or the muscles or skin of his neck. These movements would naturally have offered greater disturbance to the temperatures indicated. At the same instant, a second thermocouple was applied to the skin of the neck of the same subject at a point about an inch distant from the first couple. The junction and leads of this couple were drawn more tightly so that the entire cross-section of the wire and junction was depressed approximately one-half millimeter below the normal surface. This couple indicated a skin temperature approximately one-half degree higher than the first couple and came to equilibrium in approximately the same time.

After couple No. 2 had reached equilibrium with the skin, it was placed

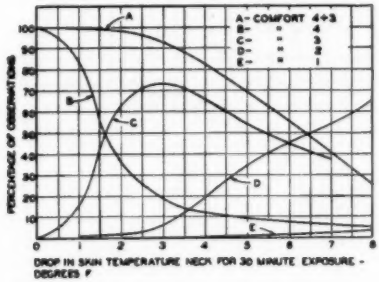

FIG. 11. RELATION BETWEEN DROP IN SKIN TEMPERATURE OF THE NECK AND PER CENT OF DIFFERENT DEGREES OF COMFORT

in the air long enough to become in equilibrium with it at an elapsed time of 17 min. Then it was again applied to the skin of the neck just tight enough to make sure that it was in contact. A little greater time is indicated for equilibrium to be reached, and a temperature about one-half degree lower is indicated. These three applications of the bare couple indicate variations which may be obtained under such conditions. One cannot, with certainty, assume that any one of the three is accurate. For all observations here reported, the Laboratory used the couple applied with medium tightness. The couples were made of No. 36 B.&S. gage copper-constantan wires, double silk covered. The wires were bare of insulation for a distance of approximately ⅛-in. either side of the junction. Beyond this distance, the insulation was firmly attached to the wire with wax.

At an elapsed time of 31½ min, couple No. 2 was attached to a cork ½ in. in diameter and about ⅜-in. thick. While both couple and cork were in equilibrium with the air, they were placed against the neck. Although the indicated temperature rose rapidly, it did not reach equilibrium until 10 or

12 min had elapsed. The indicated temperature was from 1 to 1¼ deg higher than that indicated by the bare couple applied with medium tightness. The reason for the greater lag is obviously due to the greater heat capacity of the cork and its initial temperature. The reason for the continued rise in the indicated temperature above that given by the bare couple was undoubtedly the interference with normal heat dissipation by the covered portion of the skin.

Observations of skin temperature with different methods of application indicate that a greater lag and a slightly higher temperature are obtained with a cork covered couple. On the other hand, a somewhat more constant temperature is indicated. Undoubtedly normal movements of the subject and chance variations in method of application have less effect. In some types of work ease of making observations and constancy of results may be more important than a small error in the indicated temperature. Furthermore, it should be

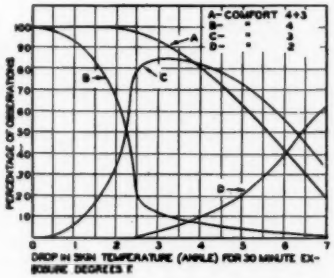

FIG. 12. RELATION BETWEEN DROP IN SKIN TEMPERATURE OF THE ANKLE AND PER CENT OF DIFFERENT DEGREES OF COMFORT

observed that the thermocouple applied with cork was previously in equilibrium with the air at a considerably lower temperature. A more satisfactory performance would result if, while the instrument is not in use, it were kept in equilibrium with the temperature not widely different from the skin surface to be measured.

On the other hand, the bare couple, when properly applied, undoubtedly gives more nearly the true temperature. Much greater care must be exercised in applying it in order to avoid breakage and, if consistent results are desired, the junction should always be applied with the same degree of tightness. These differences would indicate that the laboratory practice of applying bare couples gives better results for research where the true temperature and its effect on heat transfer are important, and where greater skill on the part of the observer can be exercised in making the application. For routine observations, in places where variations in skin temperature of some considerable magnitude are the only objectives, the use of thermocouples applied to a cork pad may be equally satisfactory and more convenient.

NORMAL SKIN TEMPERATURES

There has been some speculation and difference of opinion concerning the normal skin temperature of persons under different conditions of activity and in different atmospheric conditions. While this was not the purpose of the investigation, the many observations of skin temperature in equilibrium with given atmospheric conditions make it possible to shed additional light on this subject which may be of some interest. Figs. 5 and 6 give skin temperatures

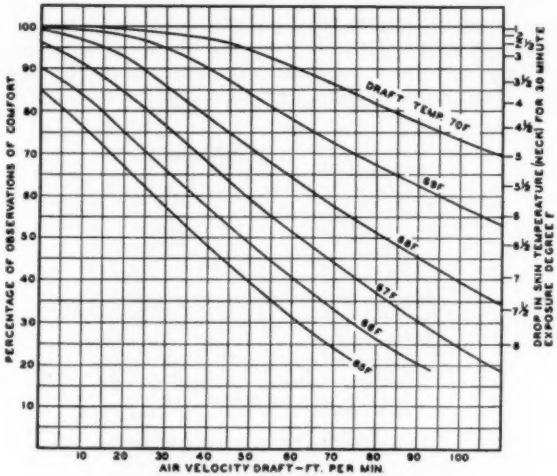

FIG. 13. RELATION BETWEEN AIR VELOCITY (DRAFT) AND PER CENT OF OBSERVATIONS OF COMFORT (LEFT HAND SCALE) AND DROP IN NECK SKIN TEMPERATURE (RIGHT HAND SCALE) AFTER 30 MIN EXPOSURE FOR DIFFERENT DRAFT TEMPERATURES

in equilibrium with air at different dry bulb temperatures and usually with 50 per cent relative humidity. A few observations were made at other relative humidities as indicated. The data show a spread of from 4 to 4½ deg for the skin temperature of the ankle and from 3 to 4 deg for the skin temperature of the neck at various air conditions. Curves are drawn giving approximately the limits of the spread. In all cases there is an apparent rise in skin temperature with higher dry-bulb temperature. It is probable that more intensive study of this subject would indicate curves other than straight lines, particularly in the region of change from sensible to insensible perspiration. For the ankle, neck, and abdomen, the greatest number of observations was made at 70 F dry-bulb and 50 per cent relative humidity. It is of interest to note the widely different temperatures observed for the different parts of the body. All of the observations were made on a total of 15 male persons ranging from 18 to 30

years of age. It is not the purpose of this paper to discuss the physiological significance of these observations.

DRAFTS

The observations on the effect of drafts were made on 10 male subjects ranging from 18 to 30 years. For each test the subject remained in one of

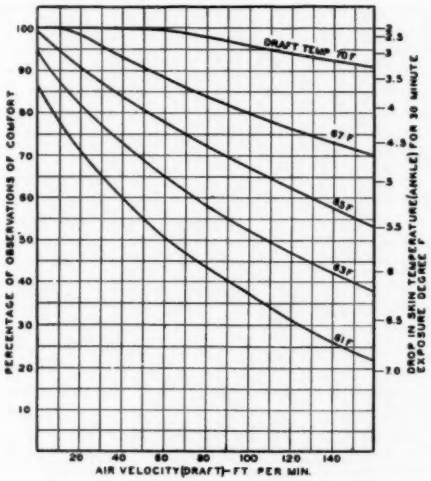
PERCENTAGE OF OBSERVATIONS OF COMFORT
DROP IN SKIN TEMPERATURE (ANKLE) FOR 30 MINUTE EXPOSURE DEGREE F
AIR VELOCITY (DRAFT) - FT PER MIN.
DRAFT TEMP 70 F
67 F
65 F
63 F
61 F

FIG. 14. RELATION BETWEEN AIR VELOCITY (DRAFT) AND PER CENT OF OBSERVATIONS OF COMFORT (LEFT HAND SCALE) AND DROP IN ANKLE SKIN TEMPERATURE (RIGHT HAND SCALE) AFTER 30 MIN EXPOSURE FOR DIFFERENT DRAFT TEMPERATURES

the psychrometric chambers, kept at the desired constant temperature and humidity, for at least a half-hour before observations were made.

In making the study it soon became apparent that more constant results could be obtained by observing the indicated change in skin temperature of any part of the body from a condition having substantially zero velocity, rather than from the normal air velocity pertaining in the psychrometric room. This was because of the small variations in air velocities in the psychrometric room set up by movements of observers and other uncontrolled factors which gave somewhat variable skin temperatures. Conditions of practically zero velocity were realized, for neck observations, by seating the subject, during the one-half hour preliminary period, with his neck close to the hood through which the impinging draft condition was later applied. No air movement was supplied in

this period. Thus, undoubtedly, a normal skin temperature, lower than that which would have prevailed if this hood had not been used, was obtained.

Considerable difficulty was experienced in the preliminary study in getting consistent results on the ankle until it was observed that, almost without exception, there was a drop in ankle skin temperature after the subject sat down in the psychrometric room. This was corrected by applying thermocouples to the skin of both ankles so as to measure the change in the affected ankle from that of the ankle not affected by the draft. In order to eliminate further the

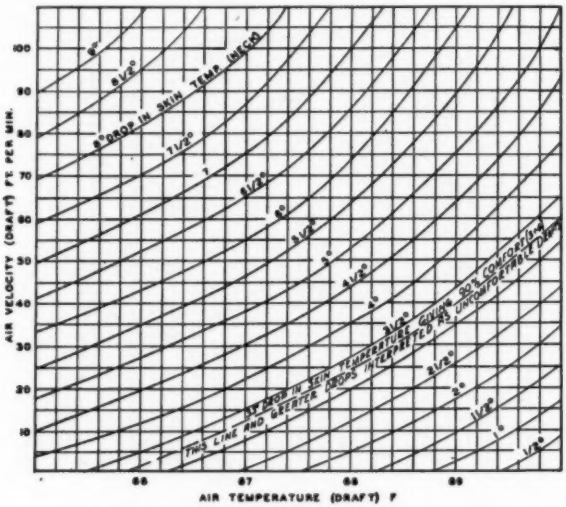

FIG. 15. RELATION BETWEEN THE TEMPERATURE AND VELOCITY OF A DRAFT AND THE DROP IN SKIN TEMPERATURE OF THE NECK. SKIN TEMPERATURE DROP OF 3.3 DEG AND GREATER INTERPRETED AS OBJECTIONABLE DRAFT

uncontrolled effect of the variable air velocity in the room, the subject's two feet were placed in separate, corrugated paper shipping boxes. Preliminary observations were made on both ankles under these conditions. At the close of the time of the preliminary observations, air of the temperature and velocity constituting the draft to be studied was directed against one of the ankles, while the other remained in the substantially still air conditions pertaining in the other box. Therefore, all drops in skin temperature and feeling of discomfort due to the draft on the affected part are given in reference to substantially still air at 70 F and 50 per cent relative humidity, rather than to the reaction in room air of normal velocity as observed by the Kata-thermometer.

Fig. 7 shows the results of a single test on the neck of a subject who had previously spent some time in the psychrometric room with the back of his neck in close proximity to the hood as indicated in Fig. 3. Prior to zero elapsed

time, the psychrometric chamber was held at 70 F dry-bulb and 50 per cent relative humidity, or 66 F effective temperature. The conditions in contact with the back of the neck were, therefore, approximately the same excepting that the air was still. Under these conditions, the neck temperature throughout the 20 min preliminary period varied very little from 91 F, until at an elapsed time of 20 min air at 68 F and a velocity of 11 fpm was directed against the back of the neck through the hood. The skin temperature of the neck dropped progressively and continuously for the next 30 min. The shape of the curve would indicate that a new equilibrium temperature would be reached somewhere

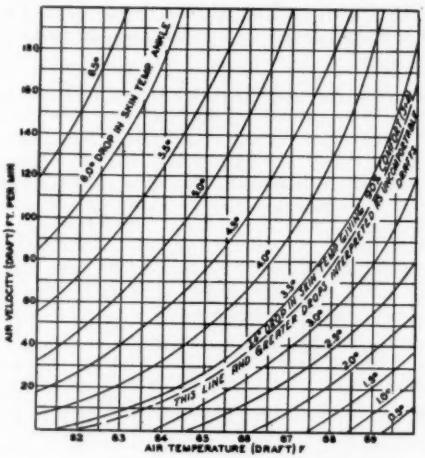

FIG. 16. RELATION BETWEEN THE TEMPERATURE AND VELOCITY OF A DRAFT AND THE DROP IN SKIN TEMPERATURE OF THE ANKLE. SKIN TEMPERATURE DROP OF 3.4 DEG AND GREATER INTERPRETED AS OBJECTIONABLE DRAFT

near 88 F. For comparison of results, the change in skin temperature during the first 30 min was used in all cases. In this neck test there was a drop in skin temperature of 2½ deg in 30 min as a result of a draft of 68 F air at 11 fpm velocity below that pertaining for 70 F air at substantially no velocity.

Seventy similar tests were made on the neck with draft temperatures ranging from 70 to 65 F and velocities ranging from 0 to 110 fpm. The drop in skin temperatures for the different tests is plotted against the draft velocity, for the different draft temperatures, in Fig. 8. While there is considerable variation in the constancy of the test points, they do indicate a rather definite series of curves which show increased drop in skin temperatures with increase in velocity and with decrease in the draft temperature.

A single test on the ankle and the results of a large series of such tests are plotted in Figs. 9 and 10, respectively. It will be observed that after 30 min exposure the affected ankle temperature dropped slightly more than 2 deg below

that of the ankle not affected by the draft. Within 40 min the drop in temperature reached over 2½ deg. The feeling of warmth indicated was *4* or *ideal* in the still air, and also in the draft until 27½ min elapsed. Within another 5 min *3* or *comfortably* cool was recorded.

The results of 71 tests on the ankle, given in Fig. 10, show about the same variations from the curves drawn, as for the neck. The two sets of curves have approximately the same characteristics, with a somewhat less drop in ankle temperature for a given draft condition than for the neck.

The relation between the drop in skin temperature and the per cent of comfort observations is plotted in Fig. 11 for the neck, and in Fig. 12 for the ankle. The same arbitrary index of comfort used in earlier laboratory studies

TABLE 1—OBJECTIONABLE DRAFT CONDITIONS BASED UPON 90 PER CENT COMFORT (*3* AND *4*)

NECK		ANKLE	
Dry-Bulb Temp. F	Velocity Fpm	Dry-Bulb Temp. F	Velocity Fpm
70	60	70	165
69	40	69	111
68	24.5	68	77
67	12.5	67	55
66	0	66	37
		65	24
		64	13
		63	6
		62	0

was applied in this investigation. According to this index *4* designates *ideally comfortable, 3—comfortably cool, 5—comfortably warm, 2—cool, 6—warm, 1—decidedly cold* and *7—hot*. Greater difficulty was experienced in getting the subjects to properly interpret and evaluate these gradations of feeling in this study than was the case in earlier work where the entire body was subjected to changes in air conditions. It was particularly hard to distinguish between feelings *3* and *4*, and it is the opinion of those connected with the study that both indications of *3* and *4* should in most instances be construed as representing conditions which are not measurably objectionable. Besides the usual curves for comfort *1, 2, 3,* and *4* in Fig. 12, there is an additional curve representing the combined votes of *3* and *4*.

The relation between the draft velocity and the percentage of comfort (*3* and *4*) for different draft temperatures applied to the neck and ankle is given in Figs. 13 and 14, respectively. Figs. 15 and 16 give the relation between

draft temperature and draft velocity for different drops in skin temperature for the neck and ankle, respectively.

If a 90 per cent indication of comfort, *3* and *4*, is accepted as the borderline between unobjectionable and objectionable drafts, then it will be noted from Figs. 11 and 12 that drops in skin temperature of 3.3 and 3.4 represent the borderline between satisfactory and unsatisfactory conditions for the neck and ankle, respectively. Thus the curve for these temperature drops in Figs. 15 and 16 (indicated by the heavy broken line) represents conditions of temperature and velocity at and above which objectionable drafts occur.

Based upon the heavy broken line curves in Figs. 15 and 16, the combinations of temperatures and velocities which constitute unsatisfactory draft conditions, are given in Table 1 for the neck and ankle. Accepting these values, it will be noted that air having a temperature of 70 F and velocity of 60 fpm directed against the neck becomes intolerable. On the other hand, if the velocity is zero, an air temperature of 66 F is equally intolerable. An air velocity of 25 fpm, as measured by a Kata-thermometer, becomes an objectionable draft when the air temperature drops to 68 F. This is in agreement with the usually accepted tolerance of $\pm$ 2 F in air conditioning practice.

The velocity of 165 fpm, which was indicated as objectionable when applied to the ankle of a subject sitting in 70 F air, seems excessively high. It should be kept in mind, however, that a person's ankle is not ordinarily subjected to 70 F air, because of the usual temperature gradient in winter air conditioned rooms. A 70 F temperature at the 3-ft level is usually associated with a floor-line temperature of 65 F. Table 1 indicates that a 65 F floor-line temperature and a 24 fpm velocity are uncomfortable, while with no air velocity a floor-line temperature of 62 F becomes equally uncomfortable.

The data given in Figs. 13 to 16 indicate a drop in skin temperature of 0.83 deg for the neck and 0.51 deg for the ankle for a 1 deg drop in dry-bulb temperature of the air when these parts of the body alone are affected by a draft. It is of interest to compare these relationships with those pertaining when the entire body is subjected to changes in temperature. From Figs. 5 and 6 the slope of the neck and ankle temperature curves indicates changes in neck and ankle skin temperatures of 0.18 and 1.17 respectively, per degree temperature change in room air affecting the entire body. At first thought it is not easy to adjust these differences. A complete knowledge of conditions would probably reconcile them on the basis that when the temperature of the air affecting the entire surface of the body is changed, the body compensates by a change in the availability of perspiration for evaporation. However, when a cooler air temperature under draft conditions affects only a limited surface area, this compensation is not provided; at least, not to the same extent.

It is of interest to note the relative effect of air temperature and air velocity in producing a given cooling effect within the temperature range of 65 to 70 F and the velocity ranges of 0 to 50 fpm. Figs. 15 and 16 show that a change in air velocity of about 15 fpm is equivalent to a change in air temperature of 1 deg.

XUM

CONCLUSIONS

1. The study shows that there is a definite relation between the drop in the skin temperature of the ankle and neck and the temperature and velocity of a draft of air striking these parts. Data on these relationships and the resulting feeling of warmth are given for room condition of approximately 70 F and 50 per cent relative humidity only. For the same change in air temperature and velocity a greater change in skin temperature is indicated for the neck.

2. This paper, accepting a recommended percentage of 90.0 for comfort indices of *3* and *4* combined, gives minimum limits in temperature and maximum limits of velocity at and beyond which intolerable drafts occur for neck and ankle.

3. A considerably higher draft velocity and lower draft temperature are indicated as possible for the ankle than for the neck. This difference is probably due to the lower temperature to which the ankle is accustomed in modern heating practice.

DISCUSSION

PROF. D. W. NELSON AND D. J. STEWART (WRITTEN): According to Mr. Houghten's Figs. 11 and 12, a drop in skin temperature, either of the neck or the ankle, of about 3.5 F is the maximum permissible. This difference appears to be constant and independent of the portion of the body, although the last sentence in the Conclusions seems to be in disagreement with the above statement. If it is true that approximately the same drop in skin temperature is uncomfortable on the neck or on the ankle, it would seem that higher velocities and/or lower draft temperatures are possible for the ankle than for the neck only because the ankle is covered and the neck exposed.

If the drop in skin temperature of 3.5 is the limit for comfort at other room temperatures than 70 F, the task of obtaining a chart like Table 1 for room temperatures other than 70 F would be a simple matter. It is, in our opinion, most desirable that this research be carried forward as rapidly as possible into the region useful in summer cooling.

In connection with the study of air distribution from side wall outlets, we felt obliged to develop a set of figures corresponding to Mr. Houghten's Table 1, but for a room temperature of 80 F. This appears as Fig. 1 in a previous paper.[1] While the two cannot be directly compared on account of the differing room temperatures, they do seem to be in general agreement.

One other minor matter appears in the next to the last paragraph before the Conclusions. It does seem most difficult to account for the great difference in skin temperature changes shown there, but it is not clear to us how those figures were obtained. It would seem from Figs. 5 and 6 that the neck and ankle skin temperature changes per degree room temperature change were about equal.

PROF. G. L. TUVE: This is an excellent paper and the work should be continued for a long time. I venture to guess that if for any strange reason we are allowed to get rid of our collars in the summer time, and can wear low collars, or if high shoes should come back, the results would be different. The year we start wearing low collars the "draft conditions" will probably seem very different from what these results show.

I would like to ask one question, and that is, just how was that thermocouple attached that you used as a reference standard?

J. R. PARSONS:[2] This matter of draft is of very serious consequence in the study that we have carried on intensively for the last four years in determining the proper

[1] Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

[2] Consulting Physicist, New York, N. Y.

effective temperature for different activities, ages and sexes. The term constantly used around the laboratory in which this work is being carried on is that air motion and draft are poison under certain conditions.

The Society has made available valuable information which has been in use right along in THE GUIDE. There have been so many complicated data accumulated on technical points that if you had the viewpoint that I have had in the last two years of confidential conferences with executives on the installation, operation and application of air conditioning, I feel that you would appreciate that the industry is in a crucial period. There are many just complaints against air conditioning.

For instance, when a large Fifth Avenue store took a vote of its book customers, a large majority voted against their putting in air conditioning due to experiences in public places.

I have been informed that one of the major baseball teams forbade athletes from riding on air conditioned trains as they did not want them to leave New York in the pink of condition and arrive in Detroit with stiff necks and charleyhorses. Now those are serious complaints and they are correctible if the proper stand is taken by this Society for the benefit of all concerned.

There is about as much relationship between the application of atmospheric conditions to nudes and clothed persons, such as we find every day, as there is between the charming young maiden and her ablutions in the bath in the morning and when we meet her dining at night with her war paint on and her full regalia. The study of both is very interesting, but the last is more practical in the way we are now considering this problem. Therefore, I make this one plea in closing, that more work be promoted of the kind that Mr. Houghten has shown in his paper,[3] on the practical application in the field and the observation of how this human animal reacts to the malpractice that is being perpetrated in most cases. But, seriously, I suggest carrying along the work to counteract that rich diet of involved data. I have been to most of these meetings, and they have been highly interesting.

M. C. BEMAN: There is no mention made in regard to any humidity conditions during these tests, and I would simply like to ask the question in regard to whether any readings were taken or if it is known what humidity was maintained during these tests.

J. J. AEBERLY: Let me state that studies of this kind are very necessary to the best interests of the membership of this Society. This intricate problem needs extensive investigation. It is true that the paper does not establish a definite relationship between drafts and man's response to them and is far from the results which we hope ultimately to obtain. My own experience with air conditioning in some 400 theaters located in a large metropolitan area indicates that general engineering data, when used as a basis for design, will invariably result in a satisfactory ventilating system, meeting the needs of the occupants in auditorium spaces, except that the engineers frequently find they have not properly provided against drafts. This condition results in much criticism being leveled at the engineer for an unsatisfactory system.

An adequate method of approach for the control of drafts in all types of ventilating systems used for comfort air conditioning is a very fundamental need.

I hope you will seriously consider this problem and encourage the men trying to solve it, to carry on the work to a definite conclusion.

MR. HOUGHTEN: There are just two questions that were asked, one by Professor Tuve as to how the thermocouples were applied that were discussed in the paper. The thermocouples in this case were applied the same as the control couple in that chart showing the different applications. We used double silk-covered thermocouple

[3] Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A.S.H.V.E. TRANSACTIONS, Vol. 44, p. 337).

wire, about number 36-gage. They were welded together for junction, and were kept bare for about ⅛-in. on either side of the junction. This junction was pulled against the skin by the two leads, and since the two parts of the body considered here, the ankle and the neck, are curved, it is quite easy to pull that junction against the surface with any degree of tightness. Now you can draw such a thing so tightly against the skin that it will be entirely imbedded in it, in which case it will undoubtedly read too high. If you apply it too loosely it will undoubtedly read too low. These couples were pulled against the skin so that the junction was halfway imbedded below the surface. You can see from that chart given in the paper that you can get a variation of possibly 1 deg depending upon how you apply such a couple or even a couple of degrees if you apply an insulated couple.

The other question raised by Mr. Beman was the relative humidity or the moisture content of the air under which these data applied. All of these data were collected in a room kept at 66 deg ET and 50 per cent relative humidity, and that gives 70 deg dry-bulb. We realized that the moisture content of the air may have some affect and undoubtedly that will have to be investigated, but to put these few data in a preliminary study on an entirely comparable basis all observations were taken under that one condition.

No. 1087

STUDY OF METHODS OF CONTROL AND TYPES OF REGISTERS AS AFFECTING TEMPERATURE VARIATIONS IN THE RESEARCH RESIDENCE

By A. P. KRATZ * AND S. KONZO ** (*MEMBERS*), URBANA, ILL.

INTRODUCTION

Preliminary Statement

IT WAS observed in the Warm Air Research Residence during the extremely severe winter heating season of 1935-1936, which was characterized by prolonged periods of cold weather, that although the air temperature at the breathing level was maintained at approximately 72 F for all weather conditions, a certain amount of discomfort was experienced in some of the first story rooms, particularly when the outdoor air temperature was lower than approximately 15 F. This condition was especially noticeable during the off-periods of the fan in the forced-air heating system.

The object of this investigation, therefore, was to study under actual service conditions some of the factors which affect the cyclical variations in temperature accompanying the operation of a forced-air heating system; and some of the factors which affect the temperature difference between the breathing level and the floor level. In addition, tests were made to determine the performance characteristics of a two-speed fan used in connection with a forced-air heating system.

The studies of the factors affecting the temperature difference between floor and breathing level have been in progress for several years in the Research Residence and have been confined to gravity warm-air, auxiliary or *booster* fan, and forced-air heating systems. These studies were supplemented with tests conducted during the winter of 1935-1936 on the forced-air heating system to determine the factors affecting the cyclical variations in room air temperatures, and with tests conducted during the winter of 1936-1937 on the two-speed fan.

Description of Research Residence

The Research Residence in Urbana, Ill., and the heating plant have been

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** Special Research Assistant Professor, Engineering Experiment Station, University of Illinois.

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1938, by S. Konzo.

completely described in a previous publication.[1] The Residence is a three-story structure of standard frame construction, in which the walls are not insulated, and no weatherstripping is used at the windows nor at the doors. With the exception of one season, it has not been equipped with storm sash.

The total space heated during these tests consisted of three rooms, a sun room, a breakfast nook, and a hallway on the first story; three rooms, a bathroom, and a hallway on the second story; and two rooms, a bathroom, and a hallway on the third story. The total volume of this heated space, from which the basement was excluded, was approximately 17,540 cu ft. The calculated heat loss was approximately 159,000 Btu per hour at an indoor-outdoor temperature difference of 80 F. The Residence is completely furnished, and during the heating season it was occupied by four people.

The heating plant consisted of a warm-air furnace used in connection with a forced-air heating system. Three cold-air returns were used which were connected into a cold-air box above the inlet to a centrifugal type of fan. The furnace was placed at the East end of the basement, and the warm-air registers were served from two main trunk systems. A more detailed description of the furnace plant, fans, and automatic controls which were used in the different tests has been included in the sections relating to the tests.

Description of Plant

During the heating seasons of 1935-1936 and 1936-1937 the furnace was of steel construction [2] specially designed for oil combustion, and was equipped with a gun-type oil burner having an oil input rate of 13 lb per hour. The centrifugal type fan, which was enclosed within the furnace casing, was equipped with forward curved blades placed on a wheel 15.5 in. in diameter. The fan was driven from the electric motor by a V-belt connection.

The control [3] of the heating plant was accomplished by means of a room thermostat operating to start and stop the combustion process and the circulating fan as shown in Fig. 1. This room thermostat was used in conjunction with two bonnet thermostats which served as high and low limit controls for the temperature of the air in the furnace bonnet. The room thermostat, which was of the heat-anticipating type, was located on an inside wall of the dining room at a height of 30-in. from the floor and was adjusted to maintain an average air temperature of approximately 72 F at the 60-in. level in all of the rooms of the Residence.

FACTORS AFFECTING TEMPERATURE VARIATION

Method of Conducting Tests

These series of tests comprised part of the routine test program conducted at the Research Residence. The air temperatures in all of the rooms of the house at the 60-in. level were maintained at approximately 72 F during all of the 24 hours which constituted one test period. Either periodic or continuous

[1] University of Illinois, *Engineering Experiment Station Bulletin* No. 266, pp. 11-17, by A. P. Kratz and S. Konzo.

[2] Performance of Oil-Fired, Warm Air Furnaces in the Research Residence, by A. P. Kratz and S. Konzo. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 215).

[3] Control Type IV described in the paper, Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, pp. 37-54).

records were made of all significant temperatures in the rooms, in the duct system, and of the outdoor air, as is completely described in previous publications.[4]

Special observations were also made of the air temperatures in the four

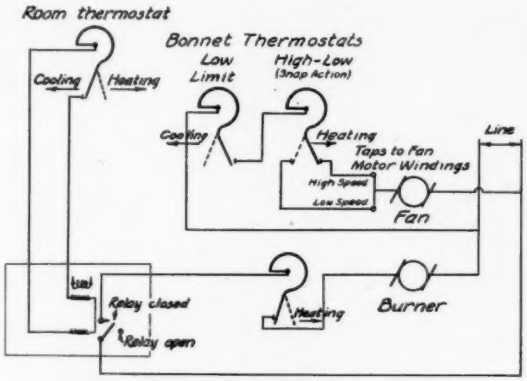

FIG. 1. SIMPLIFIED WIRING DIAGRAMS FOR TWO METHODS OF CONTROL

corners of the living room by means of thermometers placed 4-in. above the floor and 60-in. above the floor at a distance of approximately 32-in. from the adjacent walls. Thermocouples were also located in the middle of the living room at elevations of 4-in. above the floor, 60-in. above the floor, and 4-in.

[4] University of Illinois, *Engineering Experiment Station Bulletin* No. 266, by A. P. Kratz and S. Konzo; Investigation of Warm-Air Furnaces and Heating Systems, Part VI, April 10, 1934.

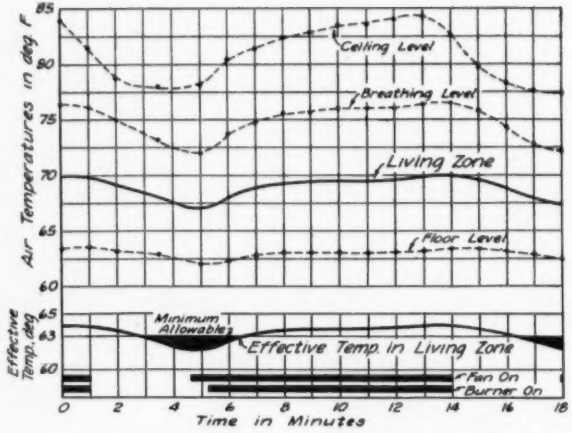

FIG. 2. (*above*) TYPICAL TEMPERATURES OF LIVING ROOM, TEST 1003, OUTDOOR = — 8 F

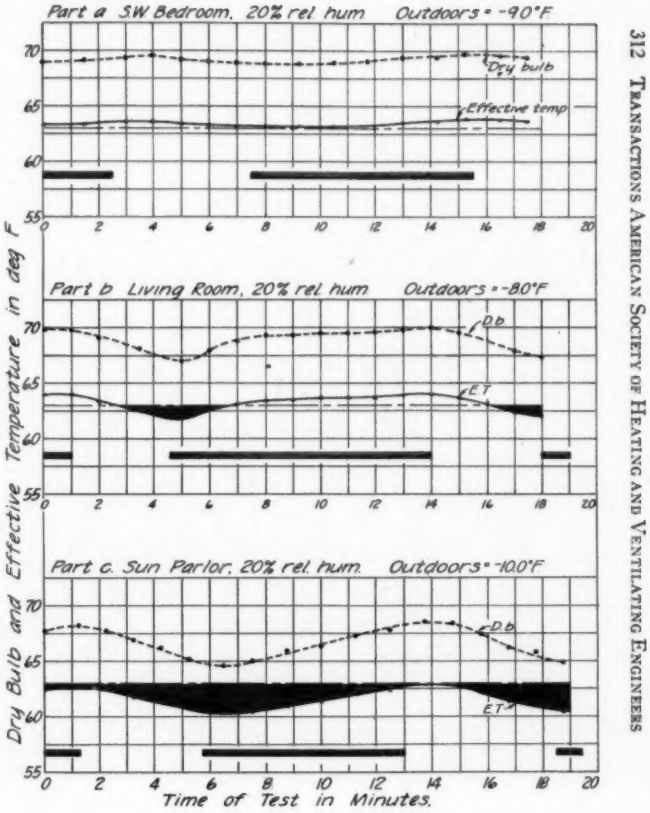
Part a SW Bedroom, 20% rel. hum. Outdoors = -90°F
Dry bulb
Effective temp
Part b Living Room, 20% rel. hum. Outdoors = -80°F
D.b.
E.T.
Part c. Sun Parlor, 20% rel. hum. Outdoors = -10.0°F
D.b.
E.T.
Dry Bulb and Effective Temperature in deg F
Time of Test in Minutes.

FIG. 3. (*right*) VARIATION IN TEMPERATURES OF LIVING ZONE IN THREE ROOMS, TEST 1003

below the ceiling. These thermocouples were used to obtain records of the variations in air temperatures at each of the three levels in this room, and were found to be more sensitive to slight changes in temperature than glass thermometers placed adjacent to them.

Results of Tests

A typical record of room air temperatures, at the floor level, breathing level, and ceiling level, during a complete cycle of the on-period and off-period of fan operation is shown in Fig. 2 for a day in which the outdoor air temperature was —8 F. This record was obtained when the fan was operated with the arrangement of thermostats shown in the lower half of Fig. 1. It may be observed that during the entire cycle of temperature variation, the air at the ceiling level was warmer than that nearer the floor level. Furthermore, when the fan operation ceased, the temperature of the air near the ceiling decreased more rapidly than did the temperature of the air near the floor.

From the standpoint of maintaining a minimum heat loss from the exposed walls in the upper portion of the room, a low temperature at the ceiling level would prove desirable. However, from the standpoint of its relation to the comfort conditions of the occupant, the temperature at the ceiling level is of very little significance. From the latter consideration, it would appear that the average of the temperatures at the floor and breathing level should be used as a criterion of the comfort conditions in the living zone. This average value, which has been designated as the *living zone* temperature, together with the relative humidity of the air in the living zone, determines the effective temperature maintained in that zone. Since a value of 63 F effective temperature has been established [5] as the minimum for comfort for persons normally clothed and surrounded by walls at the same temperature as that of the room air, this value has been used as a standard for comparison in evaluating the comfort conditions maintained in the room with various methods of operation of the heating plant.

The cyclical variation in living zone temperatures in the southwest bedroom, as indicated in part (a) of Fig. 3, was very small. Furthermore, since the effective temperature in the living zone was greater than 63 deg at all times, the temperature conditions in the room may be regarded as highly satisfactory. This small variation in living zone temperature was characteristic of the conditions which were maintained in all of the rooms on the second and third stories of the Research Residence, as is indicated by the values listed in Table 1.

On the other hand, the variation in the living zone temperature for rooms on the first story was comparatively large, it being approximately three times that of the rooms on the second and third stories. This difference in the temperature variations between first story rooms and rooms on the second and third stories resulted from the *gravity* action accompanying the intermittent operation of the fan. The anemometer readings obtained at the face of the warm air registers indicated that during the off-periods of the fan practically no gravity circulation occurred in the ducts to the first story rooms. Hence, heated air was delivered to the rooms intermittently, and only during periods of fan operation. On the other hand, very marked gravity circulation oc-

[5] Comfort Chart of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS (A.S.H.V.E. GUIDE, 1937, page 68, Fig. 6).

curred in the ducts to the second and third stories, the amount of air delivered being determined by the resistance to gravity flow of the particular ducts. Heated air, therefore, was being delivered to these rooms continuously, although in varying amounts.

The variation in living zone temperature in the living room was also representative of the conditions existing in the kitchen and hallway on the first story, and is illustrated by the curves shown in part (b) of Fig. 3. It may be noted from the shaded section in the curve for effective temperatures, that approximately 2 min after the circulating fan was stopped the effective temperature in the living zone was reduced to less than 63 deg. The comfort

TABLE 1—CHANGE IN LIVING ZONE TEMPERATURE DURING ON-PERIODS AND OFF-PERIODS OF FAN[a]

(Difference between maximum and minimum values)

STORY	ROOM	CHANGE IN TEMPERATURE, F	AVERAGE FOR STORY, F
First	Sun Room	3.8	...
	Dining Room	3.5	...
	Hall	2.0	...
	Living Room	2.2	...
	Kitchen	1.8	2.7
Second	E. Bedroom	0.2	...
	S. W. Bedroom	0.9	...
	N. W. Bedroom	1.5	...
	Bath	0.7	0.8
Third	W. Dormitory	1.1	...
	E. Dormitory	0.5	0.8

[a] Temperature of living zone determined by average of air temperatures at levels 4 in. and 60 in. above floor.
Test made on February 18, 1936. Test No. 1003. Series 1–35 Controls.
Outdoor temperature during test period approximately − 6 F.

conditions cannot, therefore, be regarded as entirely satisfactory, although either a slight increase in the average living zone temperature or a more frequent cyclical operation of the fan would be sufficient to provide satisfactory operation at all times.

The temperature conditions maintained in the sun room, as indicated by the curves shown in part (c) of Fig. 3, were not satisfactory, since the effective temperature in the living zone did not attain a value of 63 deg at any time. The variation of 3.8 F in the living zone temperature of the sun room was larger than that in any other room in the house, although it was only slightly larger than that of the adjacent dining room, which was affected to some extent by the temperatures in the sun room.

The conditions obtained in the sun room during extremely cold weather and with intermittent operation of the fan are to some extent representative of the problems encountered in the case of the isolated room under similar conditions of operation. In the first place, the sun room was exposed to the outdoors on three sides and the roof, and the floor was exposed to an unheated, unexcavated space. Furthermore, the room was equipped with 125

sq ft of single-pane window surface and had a heat loss of 30,050 Btu per hour, based on an 80 F temperature difference. Since the volume of the room was 1150 cu ft, the heat loss per cubic foot of internal volume was 26.1 Btu, or approximately 2½ to 4 times that of any other room in the house. It is not surprising, therefore, that the temperature differential from the breathing level to the floor was extremely large in cold weather, resulting in a living zone temperature that was below the minimum required for comfort, even when the plant was in operation. In addition, when the fan was stopped, the sun room, due to the extremely large heat losses, tended to cool much faster than did the other rooms in the house.

The conditions which contribute towards discomfort can be alleviated, corrected, or avoided entirely by modifications in operation of the plant and by the reduction of the heat losses from the room. The following corrective measures may be suggested:

(a) In the case of an isolated room, or a room having excessive heat losses, it is important that the losses be minimized by means of weatherstripping, storm sash, sidewall insulation, ceiling insulation, or floor insulation. Such improvements in house construction not only tend to reduce the cooling rate of the room when the fan is stopped, but also tend to decrease the temperature differential from ceiling to floor.

(b) In many cases in which the exposure of different rooms is unequal it is common practice to use zone controls.

(c) It is good practice to provide in the original design of the plant for the maintenance of slightly higher air temperatures in isolated rooms, so that the average air temperature during a complete cycle of operation will be increased.

(d) Any method of operation or any device which will tend to reduce the temperature differential from breathing level to floor is particularly desirable. In fact it may be noted that with the maintenance of a uniform breathing level temperature in all the rooms of the house, any differences in the comfort conditions in the different rooms result largely from differences in the temperature gradients maintained in the rooms.

(e) The conditions of discomfort arising from the off-periods of the fan can probably be minimized by providing for greater duration of fan operation. There is some tendency in commercial practice, particularly in the case of large installations, to use continuous fan operation. In this connection it is probable that the most ideal conditions of comfort will be attained when both the fan and the burner tend to operate continuously.

(f) In the case of intermittent fan operation some improvement in conditions may be obtained by more frequent operation of the fan in order to reduce the length of each cooling period.

In this discussion some methods of operation suggested by items *d*, *e*, and *f* have been investigated, since the suggestions in items *a*, *b*, and *c* seemed to be more or less self-evident without necessity for proof.

EFFECT OF METHOD OF CONTROL ON TEMPERATURE VARIATION

Method of Conducting Tests

A preliminary series of tests was run to determine the effect of four different methods of control on the temperature variations obtained in the living zone of the living room. All observations were made when the outdoor temperature was between —2 F and +4 F, since the conditions of discomfort were noticeable only in very cold weather and were not of any consequence in average winter weather.

Results of Tests

With normal intermittent operation of the fan it was observed that during zero weather the fan operated 4.6 times per hour, and the maximum variation in the living zone temperature was 2.7 F. No observable difference in either item was obtained when the room thermostat, which was of the heat-anticipating type, was lowered from the 60-in. level to the 30-in. level on the inside wall of the dining room.

When the same room thermostat was located at the 60-in. level on the inside wall of the sun room the frequency of fan operation was increased to 5.2 times per hour, with an accompanying increase in the number of off-periods of the fan. To offset this, however, there was a decrease in the amount of variation of the living zone temperature to 2.4 F. Although the sun room cannot be regarded as the proper place for the location of a master thermostat, on account of the pronounced effects of sun exposure on that room, the tests did indicate that an increase in sensitivity of the room thermostat is desirable since the cooling period during the off-period of the fan was shortened. They also indicated that the room thermostat should preferably be placed in a room that is subject to rather wide fluctuations in temperature if sensitive operation of the instrument is desired.

All of the three series of tests just described were made with intermittent operation of the fan. In the fourth series of tests, the same room thermostat was located at the 60-in. level in the dining room, and the control instruments were so arranged that the fan operated as long as the bonnet air temperature was greater than 125 F. In all of these tests, however, the burner was operated intermittently. Due to the fact that the volume of air circulated was large, the fan did not run continuously even in zero weather. In order for the fan to operate continuously during zero weather, the air delivery would have had to be reduced considerably from the 1675 cfm delivered during the tests. In any case, the low temperature limit control in the bonnet operated so that the average frequency of fan operation was 2.1 times per hour for zero weather. This was materially less than the 4.6 times per hour obtained for normal intermittent operation of the fan. Since the fan was operated as long as the temperature in the bonnet was 125 F, or more, there was some tendency for the air temperatures in the room to over-run, and there was also a slight delay in the delivery of sufficient heat on the first demand of the thermostat. As a result the maximum variation in the living zone temperature was 3.3 F instead of the 2.7 F variation obtained with normal intermittent operation.

Summary

From a study of the four methods of control used in these tests it was evident that more frequent cyclical operation of the fan, produced by more sensitive action of the room thermostat, tended to reduce the length of the off-periods of the fan, and hence the temperature variation in the living zone. The possible improvements in this direction are to some extent limited. It was also indicated from the tests made with the fan operation limited to bonnet temperatures above 125 F that long operation of the fan necessarily tended to reduce the number of off-periods occurring during the day, but did not necessarily reduce the length of each off-period. The latter method,

however, entails greater operating costs for the fan as compared with intermittent operation with the fan controlled primarily by the room thermostat. The possible application of a two-speed fan control which would retain the desirable features of long operation without the accompanying added fan cost was indicated from these studies and such tests were made during the following season.

PERFORMANCE CHARACTERISTICS WITH A TWO-SPEED FAN

Preliminary Statement

In the design of a duct system for a forced-air heating plant the assumption is tacitly made that the operation of the fan will be continuous. Although continuous operation of the fan is commonly used in practice for large heating installations, in a large majority of the smaller installations the greater electrical cost accompanying continuous operation has resulted in the extensive use of an intermittently operated fan controlled from a master room thermostat. In certain types of large installations and under extreme weather conditions the latter method of fan operation may not prove to be entirely satisfactory, as has been indicated in the preceding section. Hence a method of operation which combines long operating periods for the fan with reasonable electrical costs is worthy of investigation. Such an investigation of the performance of a two-speed fan in a forced-air heating system was made during the heating season of 1936-1937. It should be noted that the purpose in using two speeds for the fan was to vary the volume of air delivered, and that this could also be accomplished by using dampers to control the output of a single-speed fan. The results obtained were compared with those obtained with the usual single-speed, intermittently operated fan.

Method of Conducting Tests

In the tests made with a single-speed fan (Series 2-36), the speed of the fan was adjusted by means of a variable diameter pulley so that the air delivery was maintained at 1675 cfm. In the tests made with the two-speed fan (Series 1-36), the same high speed was retained so that an air delivery of 1675 cfm was obtained, and a low speed was selected such that an air delivery of 1000 cfm was obtained. In this particular installation it was not possible to obtain a lower fan speed and hence a lower air delivery. It is of interest to note that this air delivery at low speed, amounting to 1000 cfm, or approximately 60 per cent of the delivery at high speed, was materially higher than the normal air deliveries obtained with the gravity warm-air furnace in the Research Residence. In the latter installation, air deliveries ranging from approximately 500 cfm to 800 cfm were reported.[6]

In the case of the tests with the single-speed fan operated intermittently, the control [7] of the heating plant was accomplished by means of a room thermostat operating to start and to stop the combustion process and the circulating fan as shown in Fig. 1. This room thermostat was used in conjunction with two bonnet thermostats, one of which served to stop the fan when the bonnet

[6] University of Illinois, *Engineering Experiment Station Bulletin* No. 189, by A. C. Willard, A. P. Kratz, and V. S. Day, Investigation of Warm-Air Furnaces and Heating Systems, Part IV, January 8, 1929, p. 56.

[7] Loc. Cit. See Note 3.

air temperature became less than 125 F, and the other of which served to stop the burner when the bonnet air temperature became greater than 175 F. The room thermostat, which was of the heat-anticipating type, was located on an inside wall of the dining room at a height of 30 in. from the floor and was adjusted to maintain an average air temperature of approximately 72 F at the 60-in. level in all of the rooms of the Residence.

In the case of the tests made with the two-speed fan the room thermostat was not changed, but modifications were made in the electrical connections for the bonnet thermostats and also in the settings of the bonnet thermostats. The control of the heating plant was accomplished by means of the room thermostat operating to start and stop the combustion process only as shown in Fig. 1. When the bonnet air temperature was less than 125 F the fan did not operate; when the bonnet air temperature was between 125 F and 160 F the fan was operated at low speed; and when the bonnet air temperature exceeded 160 F the fan was operated at high speed. In both series of tests the same control instruments were used in order to eliminate any effects due to differences in the sensitivity and setting of the instruments; and the same electric motor (½ hp capacitor type, 110 volts, 60 cycle 1140 rpm maximum speed) was used for the fan in order to eliminate effects due to differences in motor characteristics.

Either periodic or continuous records were made of all significant temperatures in the rooms and in the duct system as explained in the previous section. In addition to the temperature records, complete data were obtained for each 24-hour test period on the fuel consumption, the total integrated time of operation of the fan and of the burner, the total electrical energy consumption of the fan and of the burner, and the total number of both the off-periods and the on-periods of the circulating fan.

Limitation of Results

It should be recognized that the results obtained in these tests were influenced to a large extent by the conditions which were imposed and by the performance of the equipment used. For example, the selection of 1000 cfm delivery for low speed operation of the two-speed fan was quite arbitrary, and the results obtained would have been modified slightly if a delivery either greater or less than this had been used. Also the relative power costs for a two-speed fan motor and a single-speed fan motor would be dependent on the special characteristics of the motor used. These restrictions and limitations which apply to practically all tests made under similar service conditions do not, however, invalidate the results obtained, provided that the operating conditions do not deviate too far from those which would be considered as normal in ordinary applications.

Fuel Consumption

The operating conditions which were maintained for the oil burner and the oil-burning furnace were identical in the two series of tests. The oil input rate was approximately 13 lb per hour, the air supply to the burner was adjusted to maintain 9.5 per cent of CO_2 in the flue gas, and the draft in the combustion chamber was maintained at 0.02 in. of water gage. For both

series of tests the data on fuel consumption were obtained for a wide range of weather conditions, and the consumption of fuel oil in pounds per 24 hours has been plotted against the difference in temperature existing between the indoors and the outdoors for the same period, as shown in Fig. 4. The plotted points deviate to some extent from the curve representing the average of the observed data, and these deviations can be attributed to the wind and sun

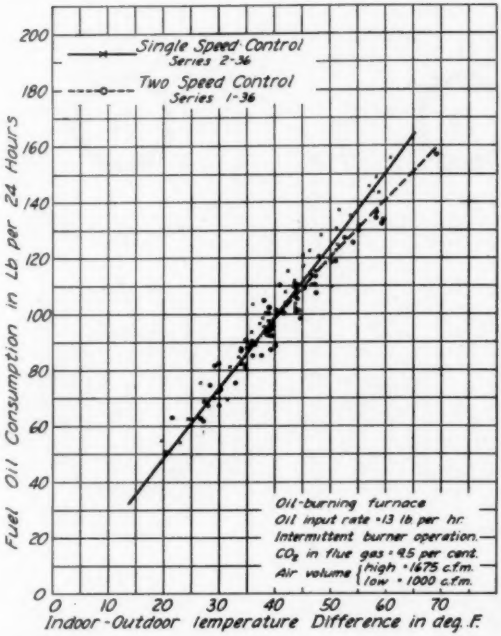
Single Speed Control
Series 2-36
Two Speed Control
Series 1-36
Fuel Oil Consumption in Lb per 24 Hours
Indoor-Outdoor Temperature Difference in deg. F.
Oil-burning furnace
Oil input rate = 13 lb per hr.
Intermittent burner operation.
CO2 in flue gas = 9.5 per cent.
Air volume high = 1675 c.f.m.
low = 1000 c.f.m.

FIG. 4. FUEL OIL CONSUMPTION FOR TWO METHODS OF CONTROL

effects which cannot be represented on a curve based on temperature difference alone.

It may be noted from Fig. 4 that the full line represents the data obtained with the single-speed fan, while the broken line represents that obtained with the two-speed fan. For milder weather conditions, or for indoor-outdoor differences less than 45 F, very little difference in the fuel consumption resulted with the two methods of operation. For colder weather, or for indoor-outdoor differences greater than 45 F, the fuel consumption with the two-speed operation was slightly less than that with single-speed intermittent operation. This reduction in fuel consumption varied from 3 per cent to 8 per cent for indoor-outdoor differences varying from 50 F to 65 F. The reduction in fuel consumption obtained with lower fan speeds cannot be attributed to higher bonnet efficiencies maintained during the periods of burner operation,

since previous tests [8] had indicated that the higher bonnet efficiencies accompanied the larger quantities of air circulated over the heating surfaces. However, as may be noted from Fig. 5, the flue losses during the off-period of the burner were apparently lower with the two-speed fan than they were with the single-speed fan. In the latter case the fan was stopped at the same time that

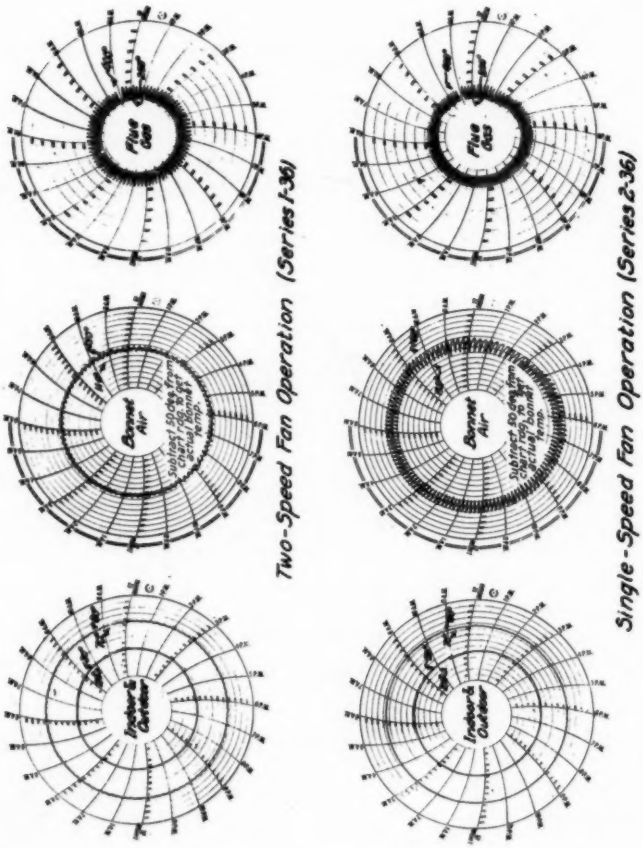

FIG. 5. TEMPERATURE RECORDS FOR TWO METHODS OF CONTROL

the burner was stopped, and the reserve heat in the furnace tended to raise the temperature of the air in the bonnet. However, in the case of the two-speed fan, the fan continued to operate for some time after the burner had stopped and the residual heat in the furnace was completely transferred from the bonnet to the rooms. It is probable, therefore, that with two-speed fan operation the reduction in the loss occurring in the hot gases passing up the

[8] Loc. Cit. See Note 2.

chimney during the off-period of the burner was equal to or slightly greater than the slight increase in these losses during the actual combustion process. Since the speed of the fan was low and the amount of residual heat was limited, the temperature over-run that accompanied these comparatively long periods of fan operation after the room was satisfied was not of sufficient consequence to be objectionable. The difference in seasonal fuel consumption for the two cases would be small, and for all practical purposes it may be

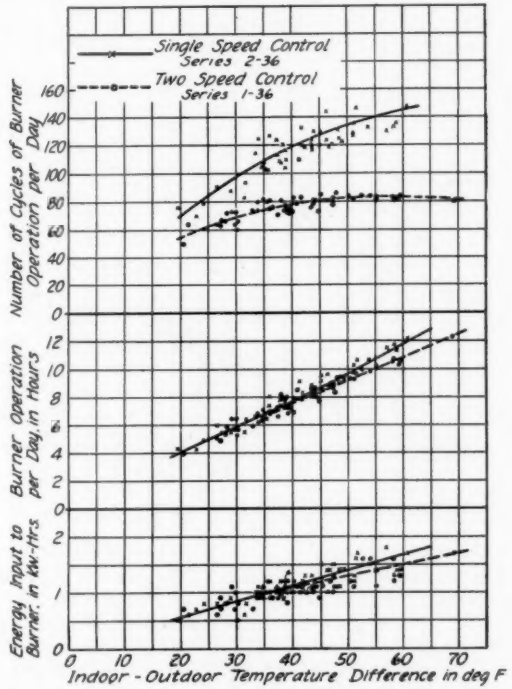
Single Speed Control
Series 2-36
Two Speed Control
Series 1-36
Number of Cycles of Burner Operation per Day
Burner Operation per Day, in Hours
Energy Input to Burner, in kw-hrs
Indoor - Outdoor Temperature Difference in deg F

FIG. 6. PERFORMANCE OF BURNER WITH TWO METHODS OF CONTROL

concluded that the fuel consumption for the two methods of operation was essentially the same.

Burner Operation

The data obtained for the operation of the burner are shown in Fig. 6. It may be noted from the top pair of curves that with low speed operation the number of cycles of operation was materially less than that with high speed operation. Since the total time of burner operation per day was approximately the same in the two cases, as shown in the middle pair of curves,

the average length of the on-period of the burner was greater with the lower fan speed. The latter may also be deduced from the flue gas temperatures shown in Fig. 5. The energy input to the burner motor was approximately the same for the two cases.

Fan Operation

Similar data for the circulating fan are shown in Fig. 7. It may be observed that marked differences were obtained in the total time of fan operation for a 24-hour period, as is also indicated by Table 2. The total running time accompanying two-speed operation was approximately 80 per cent

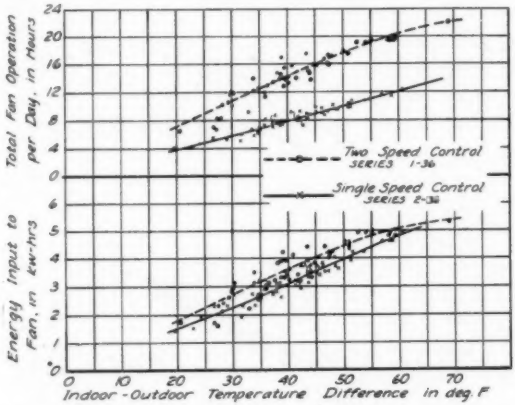
Total Fan Operation per Day, in Hours
Two Speed Control SERIES 1-36
Single Speed Control SERIES 2-36
Energy Input to Fan, in kw-hrs
Indoor-Outdoor Temperature Difference in deg. F

FIG. 7. PERFORMANCE OF FAN FOR TWO METHODS OF CONTROL

greater than that for single-speed operation of the fan. In fact for an indoor-outdoor temperature difference of 60 F the fan operated 20.6 hours out of 24, or 86 per cent of the time, when the two-speed arrangement was used.

The average power consumption of the electric motor for the fan was 0.38 kw with high speed operation and 0.25 kw with low speed operation. In other words although 0.23 kw was required per 1000 cfm delivery at high speed, as a result of slightly lower motor efficiencies at the lower speed, the rate at the low speed was 0.25 kw per 1000 cfm delivery.

TABLE 2—DATA ON OPERATION OF TWO-SPEED FAN

INDOOR-OUTDOOR TEMP. DIFFERENCE IN DEG F	TOTAL FAN OPERATION IN HOURS		INCREASE IN RUNNING TIME OF TWO-SPEED FAN OPERATION OVER SINGLE-SPEED OPERATION, PER CENT
	Single-Speed	Two-Speed	
20	3.9	7.3	87
40	7.8	14.2	82
60	12.0	20.6	72

The total energy consumption with the two-speed fan operation was from 9 to 21 per cent greater than that for single-speed operation, depending upon the severity of the weather, with an average value of approximately 15 per cent. As indicated previously these comparative values are dependent to a large extent on the performance characteristics of the motor used.

In general when the costs for fuel, for burner power, and for fan power are all taken into consideration, the total daily operating costs for the two methods of operation may be considered as practically the same.

Temperature Differential in Rooms

The results of an analysis of the average room temperature conditions maintained in all of the rooms of the Residence are shown in Fig. 8, in which the data for the differences in temperature existing between the breathing

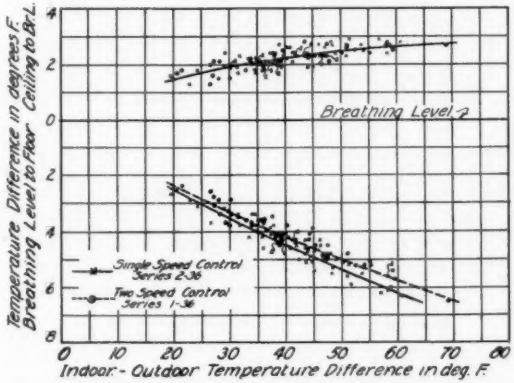

FIG. 8. ROOM TEMPERATURE DIFFERENTIAL FOR TWO METHODS OF CONTROL

level and the floor and between the ceiling and the breathing level have been plotted against the indoor-outdoor temperature difference. It may be observed that on the average, the temperature of the air near the floor with two-speed fan operation was approximately 0.4 F higher than that obtained with single-speed operation, although no appreciable difference was observed in the ceiling temperatures. Furthermore, the average value of the register air temperature was approximately 104 F with the two-speed fan owing to the preponderance of low-speed operation, and approximately 117 F with single-speed operation. It is, therefore, probable that the lower temperature of the air introduced into the room during the longer periods of low speed operation may serve as an explanation for the slightly better average room temperatures that were maintained when the two-speed fan was used.

Off-Period of Fan

In the preceding discussion the differences in the results obtained with the two methods of fan operation have been shown to be comparatively small.

However, a study of the temperature fluctuations in any given room during a complete cycle of fan operation, as well as a study of the distribution of temperatures in the different rooms, has indicated that the comfort conditions in the room were markedly improved with the two-speed operation of the fan.

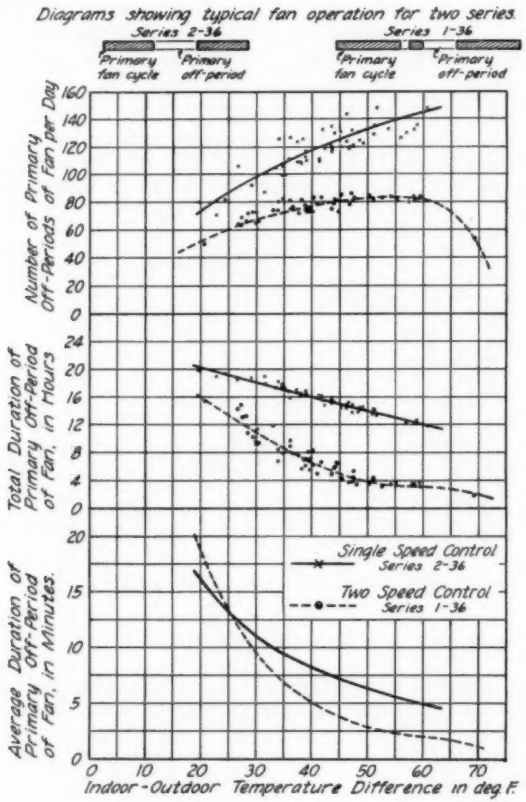
Diagrams showing typical fan operation for two series
Series 2-36
Series 1-36
Primary fan cycle
Primary off-period
Primary fan cycle
Primary off-period
Number of Primary Off-Periods of Fan per Day
Total Duration of Primary Off-Period of Fan, in Hours
Average Duration of Primary Off-Period of Fan, in Minutes.
Single Speed Control Series 2-36
Two Speed Control Series 1-36
Indoor-Outdoor Temperature Difference in deg. F.

FIG. 9. OFF-PERIOD DURATION OF FAN WITH OIL-BURNING FURNACE

In general, the conditions of discomfort occurred only during the off-periods of the fan, when no heat was being delivered to the room, and the effective temperature in the living zone had decreased to a value less than that which was considered as satisfactory. Obviously, improvements in the room comfort conditions could be effected by a reduction in the number of off-periods, in a reduction in the average duration of the off-period, or in a reduction in both factors.

The upper set of curves in Fig. 9 shows the number of primary off-periods of the fan per day. As is indicated in the inset shown at the top of the figure, the primary off-period in the case of two-speed fan operation was regarded as the longer of the two off-periods that usually follow a primary fan cycle. It is this primary off-period, rather than the very short secondary off-period, that is of significance in this study. It may be noted from the upper set of curves that the number of primary off-periods per day was much smaller for two-speed fan operation than it was for single-speed fan operation. For example, for an indoor-outdoor temperature difference of 60 F the numbers of off-periods were 82 and 145 respectively, for two-speed and single-speed operation.

The middle pair of curves shows that, in the case of two-speed operation, the total duration of time represented by the summation of the primary off-periods per day was also less than that for single-speed operation. For an indoor-outdoor temperature difference of 60 F the durations were 3.0 hours and 12 hours, respectively.

The lower pair of curves in Fig. 9 shows the average duration of the primary off-period of the fan. It may be noted that for both series of tests the average duration of the off-period diminished as the weather conditions became more severe. For an indoor-outdoor difference of 60 F the average durations of the primary off-periods were 3 min and 5 min respectively for two-speed and single-speed operation. Hence, from the standpoint of fewer number of off-periods per day and also from that of shorter duration for each off-period, it may be concluded that the heat supply to the rooms was more nearly continuous and that conditions were more favorable in the case of two-speed fan operation than in the case of single-speed operation.

Temperature Variation in Rooms

It was also observed that more nearly continuous and uninterrupted heat supply resulted in better room temperature conditions. Studies of the variations in the temperature of the air during one complete cycle of operation of the circulating fan were made in the living room. This room was 13 ft 6 in. by 20 ft and had the north, east, and south walls exposed. The supply register was located in the baseboard of the inside west wall, midway between the north and south walls. The register used in this study was 10 in. by 12 in. of the perforated grille type, and introduced the air into the room at an angle of approximately 45 deg from the horizontal. The return air grille, which was 30 in. by 8 in., was located in the floor in the north-east corner of the room.

Five preliminary tests were made including both high and low speed operation of the fan, and with the outdoor temperatures ranging between 4.5 F and 17.0 F. Each test comprised one complete cycle of fan operation and was made in order to determine the amount of deviation occurring between the temperature in the center of the room and that observed in the four corners of the room. These preliminary tests indicated that the average deviations of the corner readings from the center reading were very small; amounting to +0.02 F at the breathing level and —0.25 F at the floor. Furthermore, these deviations were not affected to any extent by the changes in the fan speed. There was some indication that the northwest corner was slightly warmer

than, and that the southeast corner was slightly colder than the other parts of the room. In general it may be concluded that the air temperatures observed in the center of a given horizontal plane in the room are representative of those prevailing at all points in the same plane in the room.

In a second set of tests the temperature variations occurring during one complete cycle of operation of the fan were investigated in the living room by means of thermocouples placed on a standard which was located in the central axis of the room. The temperature observations were made at intervals of 1 min during the complete cycle of operation. The differences between the maximum and the minimum temperatures observed at each of the three levels in the room are shown in Table 3. It may be observed that the variations in floor temperature were the same with both methods of operation, but the variations in temperature at the breathing level and ceiling were much smaller with two-speed operation than they were with single-speed operation.

In a third set of tests, which were made in the east bedroom on the second story and in the sun room on the first story, the temperature distribution between the two stories of the house was investigated. It was observed that with two-speed operation, due to the fact that the gravity circulation to the

TABLE 3—VARIATIONS IN TEMPERATURES IN LIVING ROOM[a]

FAN OPERATION	NO. OF TESTS AVERAGED	AVERAGE OUTDOOR TEMP., F	MAXIMUM TEMPERATURE VARIATION DURING ONE COMPLETE FAN CYCLE, F		
			At Floor	At Br. Level	At Ceiling
Two-Speed....	4	10.9	1.0	2.2	5.6
Single-Speed	2	11.5	1.0	3.2	8.0

[a] All observations made by means of thermocouples. Observations recorded each minute during one complete fan cycle. Tests made with no occupancy, no lights, and under fairly constant weather conditions with mild wind and very little sunshine.

second story rooms was diminished, and the total time that the fan operated was increased, the proportion of the total heat delivered to the first story rooms was slightly greater than that delivered with the single-speed operation. This in itself can be readily adjusted, but it does serve to emphasize the fact that the amount of dampering required on any given duct installation is dependent on the method of fan operation used, and is not the same for both continuous and intermittent operation of the fan.

It was indicated in a previous study that, in the case of all the second story rooms, the temperature variations were small. Observations of temperatures in the east bedroom indicated that the actual temperature gradients from breathing level to floor were the same for the two methods of fan operation, and furthermore that the maximum variation in temperature in the living zone did not exceed 0.9 F for both series. It may be concluded, therefore, that the use of the two-speed fan did not change the very favorable conditions that existed in the second story rooms when the single-speed fan was in use.

In the case of the sun room, on the first story, and for tests made with an outdoor temperature of 6 F, it was observed that the use of the two-speed fan was accompanied by an increase in the magnitude of the living zone temperature of 0.6 F and by a decrease in the maximum temperature deviation in the living zone of 0.4 F, as compared with similar values obtained with single-speed fan operation. Both tendencies indicate that more favorable conditions were maintained with the use of the two-speed fan.

Summary

The use of a two-speed fan, that operates at low speed during the greater portion of the heating season and at high speed during heavy heating demands only, requires a more expensive motor and a more complicated control system than is required for the simpler single-speed fan operated intermittently. The total operating costs, including those for oil fuel, and for power consumed by the burner motor and fan motor, were approximately the same with both methods of fan operation. In the case of the two-speed fan, however, the fan was operated a greater proportion of the day, the total number of the primary off-periods of the fan was decreased, and the average duration of the off-periods of the fan was diminished. Furthermore, the temperature conditions in the room were improved. The temperature and comfort conditions in the second story rooms, which were entirely satisfactory with single-speed fan operation, were not changed when a two-speed fan was used. In the first story rooms, not only did the two-speed operation tend to increase the magnitude of the temperatures in the living zone, but it decreased the temperature fluctuation during each fan cycle. These improvements in the comfort conditions, consisting not only of a substantial reduction in the number and lengths of off-periods of the fan per day, but also of a noticeable improvement in the variations in air temperature in first story rooms, are such that the use of a two-speed fan should be given consideration in the application of forced-air heating to small and medium sized installations. The accompanying factors of quiet operation and more moderate air movement in the rooms may also be considered as advantageous features.

FACTORS AFFECTING TEMPERATURE-DIFFERENTIAL BETWEEN BREATHING LEVEL AND FLOOR

Preliminary Statement

In the first section of this report it was indicated that the condition of discomfort experienced in some of the first story rooms was partly attributable to the difference in the temperatures that existed between the breathing level and the floor, and partly to the fluctuations in the air temperature during a cyclical operation of the fan. It was also indicated in the preceding section that the variations in the air temperatures in the rooms could be reduced to some extent by the use of a two-speed fan. In addition, improvements of equal or even greater importance in the room comfort conditions may be effected by substantial reductions in the temperature differential between breathing level and floor, as illustrated in Table 4.

For ideal conditions of operation, such as is indicated in Table 4 with no difference in temperature existing between the breathing level and the floor, the effective temperature would be 66.1 deg. Under these conditions, the living zone dry-bulb temperature might decrease at least 4 F during a cyclical operation of the fan before the effective temperature approached 63 deg. On the other hand, if the temperature differential between breathing level and floor were 6 F, the effective temperature in the living zone would be 64 deg, and a reduction in the dry-bulb temperature of only 1 F in this zone would result in an effective temperature approaching the lower allowable limit of 63 deg.

TABLE 4—EFFECTIVE TEMPERATURE IN LIVING ZONE WITH VARIOUS TEMPERATURE DIFFERENTIALS

(Based on breathing level temperature of 72 F)

AIR TEMPERATURES, F			EFFECTIVE TEMPERATURE IN LIVING ZONE,[a] DEGREES
Breathing Level	Floor Level	Average Living Zone	
72	72	72.0	66.1
72	71	71.5	65.7
72	70	71.0	65.4
72	69	70.5	65.0
72	68	70.0	64.7
72	67	69.5	64.3
72	66	69.0	64.0
72	65	68.5	63.6
72	64	68.0	63.2
72	63	67.5	62.8

[a] Effective temperature based on a relative humidity of 30 per cent. Values would be increased approximately 0.6 deg for a relative humidity of 40 per cent.

Furthermore, if the temperature differential between breathing level and floor became 8 F, the effective temperature would be 63.2 deg and any reduction in the temperature in the living zone would result in an effective temperature lower than the allowable limit. In the case of a room in which a large temperature differential exists, improvement in comfort conditions may be made either by increasing the temperature at the level of the thermostat, particularly as the weather gets colder, thus increasing the temperature at all levels; or by maintaining the same temperature at the control level and reducing the temperature differential. Since the former method is more wasteful of heat, it is apparent that the only satisfactory solution of the problem consists in the maintenance of minimum temperature differentials in the given rooms. In connection with this study an extensive series of tests has been conducted over a period of years, the results of which will be briefly reviewed.

Effect of Storm Sash

In a previous publication [9] it was shown that by equipping the windows with tightly fitting storm-sash, the room temperatures at the floor level were in-

[9] Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S. Konzo. (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, pp. 87-98).

creased, although no observable difference in the temperatures at the ceiling was recorded. The use of the additional barrier of glass against heat flow resulted in an increase in the temperature of the inner surface of the glass and in an increase in the temperature of the air which flowed down over the window sills and settled in the lower portion of the room. The data obtained in the dining room indicated that the increase in the temperature at the floor level was approximately 0.6 F for an outdoor temperature of 50 F and was approximately 1.8 F for an outdoor temperature of 0 F.

In a similar manner, the use of sidewall insulation should tend to increase the temperature of the air flowing down the wall and hence tend to increase the temperature at the floor level. Adequate protection of the windows and

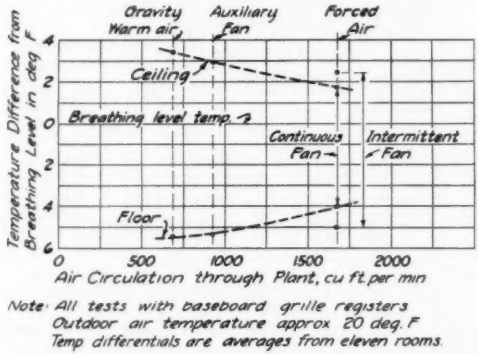

FIG. 10. TEMPERATURE DIFFERENTIALS WITH GRAVITY, AUXILIARY FAN, AND FORCED-AIR SYSTEMS

walls will, therefore, not only reduce the loss of heat from the structure, but will also promote more comfortable room conditions by increasing temperatures in the living zone, and by minimizing the heat loss from the body to cold surfaces.

Effect of Air Temperatures

Incidental to the tests made with the oil-burning furnace and with the conversion unit, it was observed that although there was no difference in the temperatures at the floor obtained with the two installations, the temperatures at the ceiling in the case of the oil-burning furnace were slightly higher than they were with the conversion unit. The same quantity of air (1675 cfm) was circulated in both cases, but the temperature of the air leaving the furnace (as measured in the north trunk duct) was approximately 4 deg higher for the oil-burning furnace than it was for the conversion unit under comparable weather conditions. This slightly warmer air entered the rooms through registers of the perforated grille type and rising to the ceiling caused slightly warmer temperatures at this level. The excess was of the order of 0.2 F and 0.8 F, respectively, for indoor-outdoor temperature differences of 30 F and 70 F.

XUM

Effect of Air Volumes

The average temperature differentials for all of the rooms in the Residence have been recorded for each of the three main types of heating systems that have been installed. The temperature differentials, obtained for the gravity warm-air, auxiliary-fan, and the forced-air system, are shown in Fig. 10. In all cases, the same type of register, consisting of a perforated grille, was used, and hence the warm air was introduced into the rooms in the same manner. In the case of the forced-air system, the differential

TABLE 5—CHANGES IN WARM AIR REGISTERS IN RESEARCH RESIDENCE[a]

Story	Room	Former Grille		New Deflecting Type Register				
						Type of Projection		Free Area[b] Per Cent
		Width In.	Height, In.	Width In.	Height, In.	Vertical Plane	Horizontal Plane	
First	Living	12	10	12	6	Straight out	Three-way deflection in all registers on the first story	Approx. 90
	Hall	14	12	14	4	Straight out		
	Dining	12	10	12	6	Straight out		
	Sun, So.	12	10	12	10	Straight out		
	Sun, No.	12	10	12	10	Straight out		
	Kitchen	14	12	14	10	Straight out		
Second	E. Bed.	12	10	12	6	Straight out	Three-way deflection	Approx. 80
	SW. Bed.	12	9	12	6	Straight out	Deflection to right	
	Bath	10	8	10	8	Straight out	Deflection to right	
	NW. Bed.	12	10	12	10	Straight out	Three-way deflection	
Third	E. Dorm.	10	8	12	10	Deflection 22 deg down	None	Approx. 86
	W. Dorm.	10	8	12	10	Deflection 22 deg down	None	

[a] All new registers selected on the basis of an average register velocity of approximately 500 fpm. All registers located in the baseboard. Change in registers made in April, 1937.

[b] Free area percentages based on opening size and not on gross size or trade designation size.

for the sun room has been included in the average, whereas it was not included in the case of the other two systems, in which the sun room was not heated. Hence, the results as shown tend slightly to favor the gravity and auxiliary-fan installations, since the inclusion of a large value for the sun room in the average would give larger differentials than those actually recorded. Even so, however, it may be noted that the smallest temperature differential was obtained with the forced-air system, particularly with continuous operation of the fan. This reduction in temperature differential may be attributed not only to the increase in the quantity of air which was circulated, but also to the decrease in the temperature of the air which was introduced into the rooms.

Effect of Register Types in East Bedroom

There has been reported [10] an extensive series of tests made to determine the effect of various types of warm-air registers on the temperature gradient in the room. These tests, which were made in the east bedroom under similar conditions of operation, indicated that for equal register air velocities, the minimum temperature differential was obtained when baseboard registers

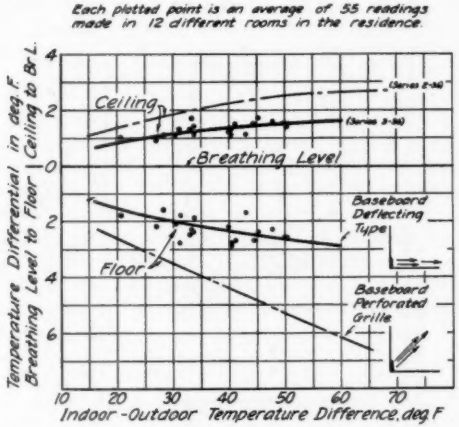

FIG. 11. ROOM TEMPERATURE DIFFERENTIALS WITH TWO TYPES OF REGISTERS (OIL-BURNING FURNACE, INTERMITTENT SINGLE-SPEED FAN)

were used containing deflectors which projected the air slightly downward and thus spread the warm air near the floor.

Effect of Deflecting Types of Register in All of the Rooms

These studies were extended, during the latter part of the 1936-1937 heating season, to include all of the 12 baseboard registers in the Residence. The large perforated grille types of register, which had been in place since the construction of the house in 1924 were replaced with smaller, deflecting type registers, the sizes of which are given in Table 5. The new registers were of such construction that the air was projected either horizontally or downward into the room near the floor level and in some cases also deflected sidewise. In the bathroom and southwest bedroom on the second story, the louvers in the registers were adjusted so that the air was deflected away from walls a distance of 3 ft in front of the registers. In the case of the bathroom and the two dormitory rooms, in which the register was located 3 in. above the baseboard, the air was projected downwards towards the floor.

[10] University of Illinois, *Engineering Experiment Station Bulletin* No. 266, by A. P. Kratz and S. Konzo.

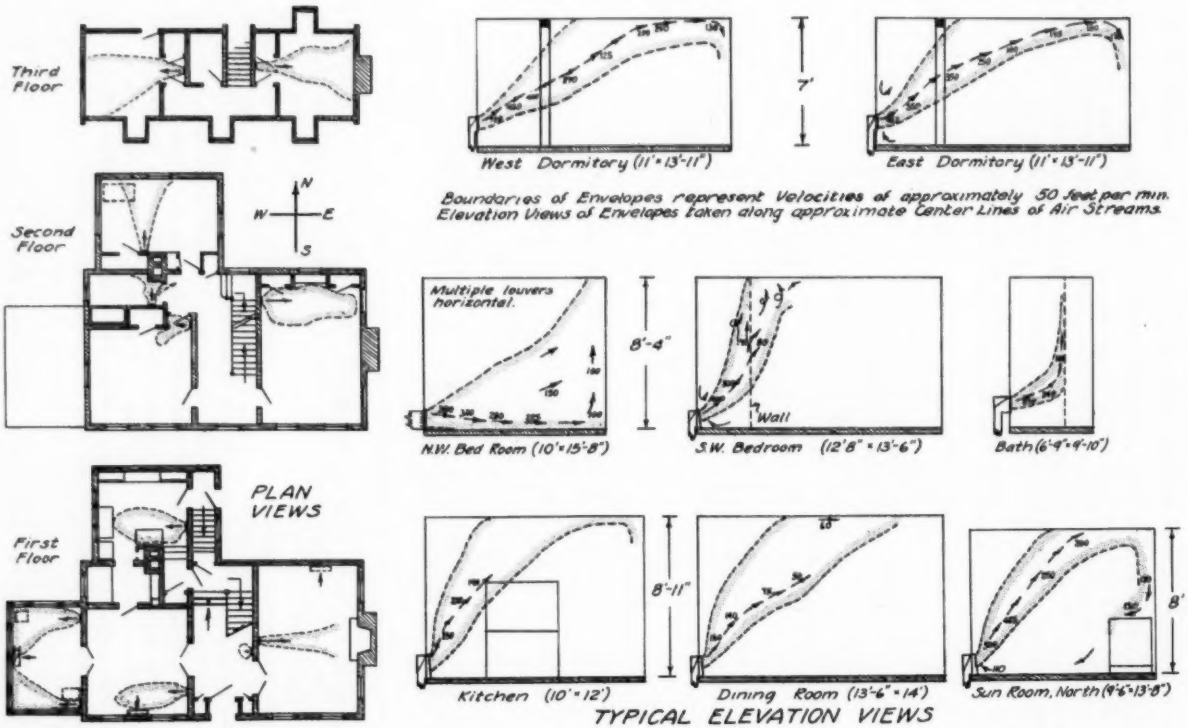

FIG. 12. PLAN AND TYPICAL ELEVATION VIEWS SHOWING AIR VELOCITY DISTRIBUTION IN ROOMS, BASEBOARD PERFORATED GRILLE REGISTERS

As in all studies on temperature differentials the average temperatures in all of the rooms at the ceiling level, breathing level, and floor level were obtained for each day, over a wide range of weather conditions. The results from the tests, which were made under conditions of operation identical with those for single-speed intermittent operation of the fan in the oil-burning furnace unit, have been plotted in Fig. 11. For purposes of comparison the temperature differential curves shown in Fig. 8 have been transferred, and are shown as broken line curves in Fig. 11. It may be noted that a very appreciable improvement in the room temperature conditions was obtained with the new registers. For an indoor-outdoor temperature difference of 50 F, for example, the total temperature differential from floor to ceiling has been reduced from 7.8 F to 4.1 F.

Furthermore, the greatest improvement was observed in the differential existing from the floor to the breathing level. For an indoor-outdoor difference of 60 F, the temperature differential from floor to breathing level was reduced from approximately 6 F to 3 F. As may be observed from Table 4 this improvement in the living zone temperature was equivalent to an increase in the effective temperature of 1 deg. This improvement in the temperature differential, particularly in the living zone, represents the largest improvement effected by any single factor which has been studied, and substantiates the conclusions which were obtained from the earlier tests made in the east bedroom alone.

Air Flow Conditions in Rooms

The distribution and magnitude of the air flow into each of the 12 rooms, before and after the registers were changed, were studied by means of anemometers and smoke in order to determine the shape and location of the envelope of the air stream. In every case in which the perforated grille type of register was used, the air was projected into the room at an angle of from approximately 45 to 60 deg and the warm air rose directly to the ceiling. In addition the air stream was confined to a small portion of the room, as shown by the envelopes in Fig. 12, and in this way local effects were accentuated. After the registers were changed the envelopes, as shown in Fig. 13, were spread out in a horizontal direction and confined to within about 10 in. above the floor; thus both diminishing any local effects and confining them to a region in which they would not be objectionable.

Summary

There remains little doubt that, by the proper application of deflecting types of register, some of the problems arising from the variation in room temperatures may be avoided. Their application should be given especial consideration in houses that are not well protected against heat loss, in rooms that have excessive exposure, in isolated rooms, and in rooms that are located on the first story.

GENERAL SUMMARY AND CONCLUSIONS

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

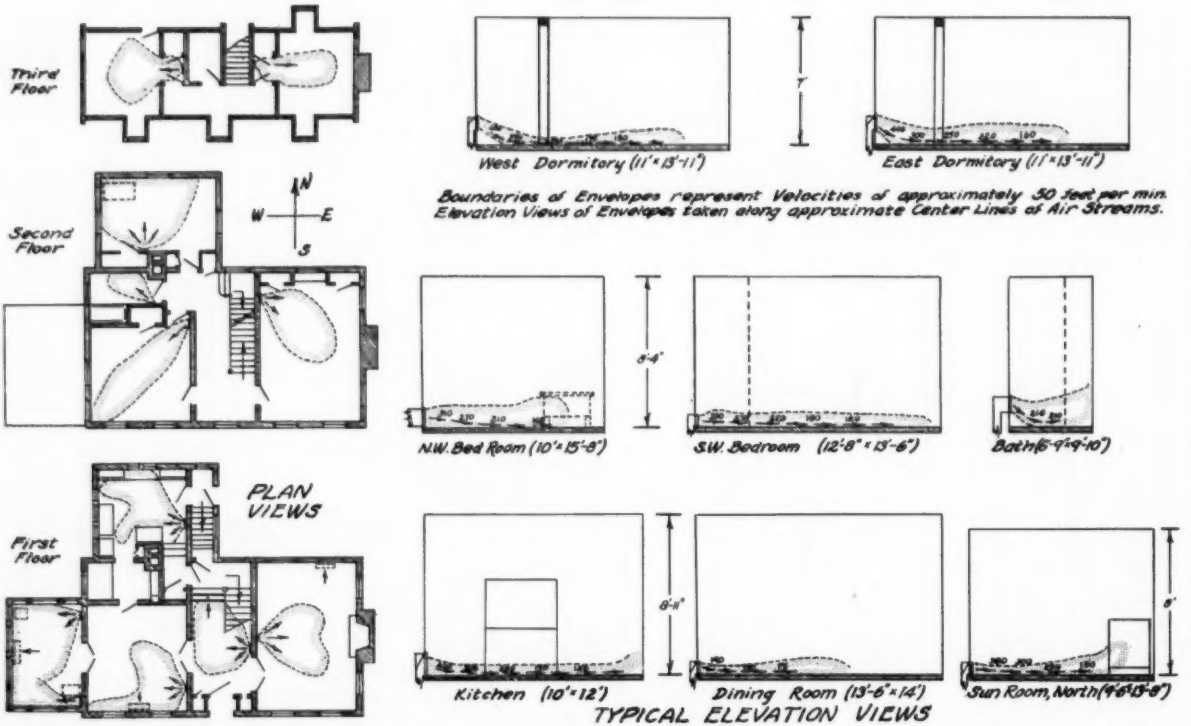

FIG. 13. PLAN AND TYPICAL ELEVATION VIEWS SHOWING AIR VELOCITY DISTRIBUTION IN ROOMS, BASEBOARD DEFLECTION REGISTERS

(1) It was observed that although the air temperature at the breathing level was maintained at approximately 72 F for all weather conditions, a certain amount of discomfort was experienced in some of the first story rooms, particularly when the outdoor air temperature was lower than approximately 15 F. This condition was especially noticeable during the off-periods of the fan in the forced-air heating system.

(2) The cyclical variation in temperature, during the on-periods and off-periods of the fan, in the living zone of the second and third story rooms was quite small, whereas that in first story rooms was about three times as large.

(3) More frequent cyclical operation of the fan tended to reduce the length of the off-periods of the fan and hence the temperature variation in the living zone. Furthermore, the number of off-periods occurring during the day was decreased when a method of control was used which provided for practically continuous fan operation.

(4) A comparison of the results of the tests made with a single-speed fan and those made with a two-speed fan indicated that the total daily costs for the two methods of operation may be considered as practically the same. However, from the standpoint of fewer number of off-periods per day and also from that of shorter duration for each off-period, it may be concluded that the heat supply to the rooms was more nearly continuous and that conditions were more favorable when the two-speed rather than the single-speed fan was used.

(5) The amount of dampering required on any given duct installation is dependent on the method of fan operation used, and is not the same for both continuous and intermittent operation of the fan.

(6) A reduction in the temperature differential between the breathing level and the floor may be effected by the use of storm sash, by the use of large volumes of circulating air at lower temperatures, and by the use of registers which deflect the air towards the floor. The results obtained from the use of deflecting types of register located in the baseboard were better than those obtained with the perforated grille type of register in the same location.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Residence,[11] at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgments are also due to: R. B. Engdahl, research graduate assistant, and to J. S. Cunningham, former research assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

[11] The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm-Air Heating and Air Conditioning Association* in December, 1924.

DISCUSSION

PROF. A. P. KRATZ: I should like to point out that while we have referred to this as a two-speed fan operation, the crux of the matter is that instead of using one large volume of air we used two volumes of air, one volume approximately half of the other. This can be accomplished either by the use of a two-speed fan or it can be accomplished by a single-speed fan and a system of damper arrangements. The end in view is to change the air volume and it can be done in either way.

No. 1088

SUMMER COOLING REQUIREMENTS OF 275 WORKERS IN AN AIR CONDITIONED OFFICE

By A. B. NEWTON * (*NON-MEMBER*), F. C. HOUGHTEN ** (*MEMBER*), CARL GUTBERLET *** AND R. W. QUALLEY † (*NON-MEMBERS*)

This paper reports the results of research, sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Committee on Research, conducted under the supervision of the Technical Advisory Committee on Comfort Air Conditioning

COOLING requirements for summer air conditioning have been the subject of extended research by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in Pittsburgh, and by the Laboratory in cooperation with the Ontario Research Foundation of Toronto, and the Texas Agricultural and Mechanical College. Two papers,[1, 2] giving the results of these studies have been published, indicating the requirements of certain definite effective temperatures with considerable variation in relative humidity. Optimum effective temperatures of 73 deg were indicated for Pittsburgh and Texas, and an effective temperature of about two degrees lower for Toronto.

A check-up on these variations has been thought desirable due to the fact that the previous studies had been made with a few young college students, trained for the work, rather than with persons of normal activities, age variations, and of both sexes. These requirements were satisfied by the excellent facilities for an extended study which were offered by a company whose general offices in Minneapolis are air conditioned and are occupied by 275 employees. The admirable arrangements for carrying on the desired survey included the following set-up and facilities:

(1) The office personnel consists of approximately 275 employees, including both men and women, having an age distribution of from slightly below 20 to 70 years,

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[1] Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215.)

[2] Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145.)

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1938, by A. B. Newton.

and other characteristics normally met with. A majority of these employees are between 20 and 40 years of age and about 55 per cent of the entire group are men.

(2) The offices are completely air-conditioned, and served by three separate systems. The control and recording equipment is unusually complete, each system having its own set of year-round automatic controls.

(3) Since the offices are under automatic temperature control from morning till night, there appeared to be an excellent opportunity to obtain data regarding the *shock effect* experienced by people when entering and the reaction experienced by people leaving summer cooled and air conditioned space.

(4) Minneapolis normally experiences one or two periods of hot weather during an average season. By starting early in the cooling season and continuing through this season until some degree of heating was required, it was expected that a study of the seasonal change in comfort requirements could be made.

BULLETIN No. 1. Subject: Air Conditioning Tests

The rapid growth of air conditioning is bringing our company an increasing amount of business, thus improving your condition of employment. At the same time it has imposed a tremendous and diversified responsibility upon all manufacturers of air conditioning apparatus including the related controls, and upon the engineering societies which are called upon to recommend proper conditions for human health and comfort.

The American Society of Heating and Ventilating Engineers has carried on research in Pittsburgh, Toronto, and Texas in an endeavor to determine the temperature and relative humidity conditions at which the largest percentage of people are comfortable and healthy during various times of the year. This work is of tremendous importance, since it provides the data necessary to make people comfortable and thus enhance the public acceptance of air conditioning.

Our company now has an opportunity to make an outstanding contribution to this field of knowledge. Because of the large group of persons employed in our air conditioned offices, and because of the complete control and recording facilities in the air conditioning equipment of these offices, we have been asked to cooperate in this work with the American Society of Heating and Ventilating Engineers by securing the comfort reactions of our office group.

We are, therefore, asking you to fill out a questionnaire, either daily or at other specified intervals. These questionnaires will be distributed and collected by some one designated for this task. Your recorded reactions will enable us to tabulate and analyse the group feelings. We cannot stress too highly that your personal unbiased feelings of warmth or chilliness are what is needed to make the work of greatest value.

Your cooperation in this work will be a distinct contribution to the necessary knowledge for more satisfactory air conditioning.

In research of this type, wherein the reactions of a large group of people are involved, the method of initiating the work and approaching the subjects to acquaint them with the problems involved is of considerable importance. Realizing that many of the persons whose reactions were desired had no personal interest in air conditioning, it was thought that the actual work of collecting data should be preceded by bulletins acquainting them with the purpose of the study. The first bulletin (reproduced herewith), sent to all employees of the

company, was intended to produce an awareness on the part of the entire office personnel of the need and importance of the work contemplated.

These bulletins were followed by a daily questionnaire card used for collection of the data and advice concerning the proper method of recording the individual daily reactions to the conditions in the office and the outside weather conditions. One of the cards was given to each employee each morning to be

DAILY QUESTIONNAIRE TO OFFICE PERSONNEL

Name_____

Location: *Zone 3*

Date: *May 27, 1937*

Insert in Proper Column, "W" for feeling of warmth, "C" for feeling of chilliness.

	IDEAL	MILD	SEVERE
1. Morning reaction to outdoors.	*I*		
2. Reaction upon entering office. Duration: *10 min.*		*C*	
3. Reaction upon leaving for outdoors at noon. Duration: *5 min.* Reaction upon leaving for unconditioned space at noon. Duration:		*W*	
4. Reaction upon re-entering office at noon. Duration:	*I*		
5. General reactions during working hours: Time: 10:30 A.M.	*I*		
3:00 P.M.		*W*	
3:30 P.M.	*I*		
6. Initial reaction during evening. Duration: *Approximately 10 min.*		*W*	
7. Continued reaction during evening			*W*

8. General remarks: *Sunshine was uncomfortable at 3:00 P. M.*

filled out during that day, and the following morning it was returned. One of the cards as returned is illustrated.

By placing *I* in the *Ideal* column, line No. 1, this person indicated that upon leaving home in the morning he experienced an ideal feeling of warmth. When he entered the office, the *C* in the *Mild* column, line No. 2, shows that he felt mildly chilly for 10 min. At noon he went outdoors, and, for 5 min felt mildly warm as indicated in the upper part of line No. 3. The lower part of line No. 3 would be used if he had gone to the unconditioned restaurant. When he re-

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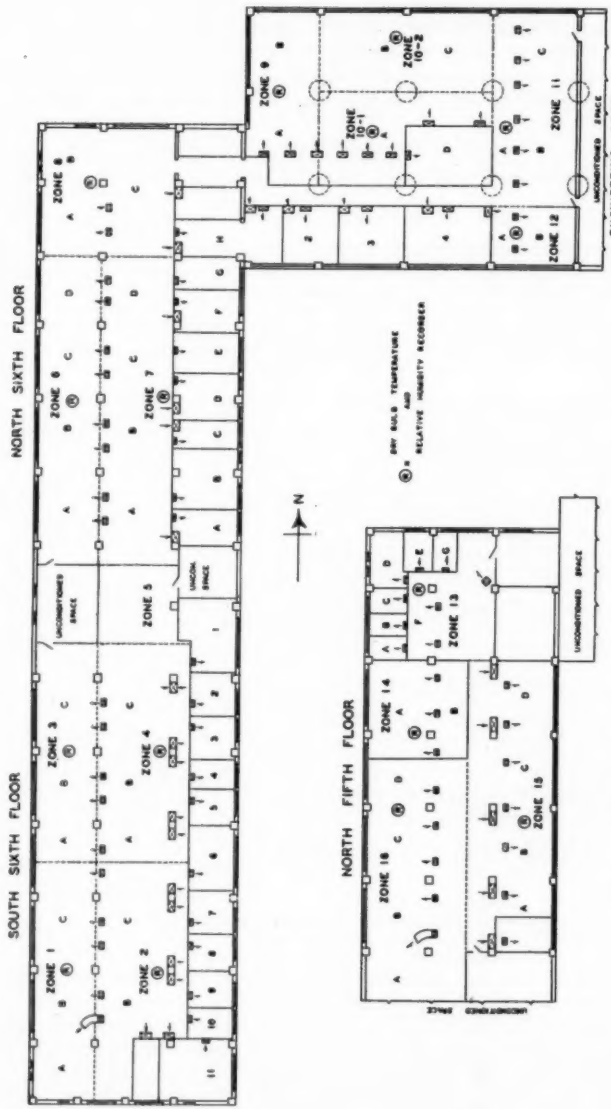

FIG. 1. AIR CONDITIONED OFFICE SPACE IN WHICH THE STUDY WAS MADE

entered the office his reaction was neither of warmth nor of chilliness, but one of ideal comfort as indicated by the *I* in line No. 4. In line No. 5 he indicates that at 10:30 A.M. he realized the office conditions were ideal, while at 3:00 P.M. he was mildly warm, as indicated by the *W* in column headed *Mild*. In line No. 6 this person has indicated that upon leaving the office at the close of work he experienced a mild feeling of warmth for a duration of 10 min. In line No. 7 he indicates by *W* in the *Severe* column that during the evening the feeling of warmth became severe. He explained under *General Remarks*, line No. 8, that his 3:00 o'clock feeling of warmth might be due to sunshine through the window.

It should be noted that the bulletins mentioned stressed the need for close cooperation and a clear expression of the feeling of those concerned. It is believed that this cooperation was obtained to a very marked degree throughout the entire test period.

DESCRIPTION OF THE CONDITIONING SYSTEMS

The conditioning apparatus supplying the offices is divided into three systems. The first of these supplies the south half of the sixth floor. The second is a zone system, supplying the north half of the fifth and sixth floors. The third system supplies the Engineering Department. The relative location of the spaces supplied by the three systems is evident in Fig. 1.

The first system consists of an air washer having a by-pass around the preheating coil, air washer sprays, and reheater coil. Provision is made for supplying outside air by means of a set of dampers located in an outdoor wall up-stream from the conditioning apparatus.

Precooling of the outdoor air in the second system is obtained by a finned surface cooling coil, located in the intake. There are dampers up-stream from this coil to provide a by-pass if necessary. The return and outdoor air is mixed in a plenum chamber up-stream from the fan. After passing through the fan, the air is divided into two streams, one of which passes over the finned surface cooling coil and the other passes over a finned surface steam coil. A water spray is supplied upstream from the steam coil for humidification when necessary. Face dampers control the amount of air passing into either stream so that reheat is readily available. The air so conditioned is then again divided into two streams, one of which goes to the north half of the sixth floor and the other to the fifth floor.

System number 3, which supplies the Engineering Department, has precooling coils for the outside air, cooling or heating of the mixture of return and outdoor air, and suitable means for by-passing around either the heating or cooling coil.

Artesian well water at 52 F is used as the cooling medium in all three systems. After passing through the cooling coils, the water may be wasted to the sewer or sprayed over the roof, depending upon outside conditions. These systems are so controlled as to provide any desired conditions within the limits of the units themselves. There is an engineer in attendance at all times to care for any emergencies that might arise. It is possible by means of centrally located recorders to observe the condition of the air at any point in its passage through any one of the systems.

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METHOD OF OBTAINING RECORDS OF PHYSICAL CONDITIONS

In a conditioned space as large as the general offices where studies were made, it was obviously necessary to group the occupants into zones in which the conditions were somewhat constant. This division was accomplished with no particular regard for occupant load by making observations of the dry-bulb temperature and relative humidity present at various times during the working day. These readings were made, one in approximately every 225 sq ft of floor space, with a standard sling psychrometer. After an analysis of this type it was relatively easy to divide the offices into zones with approximately constant conditions throughout. This division is shown in Fig. 1, the numbers referring to the zones and the letters within the zones serving to further subdivide them for a more exact location of individuals.

There were no zones which were supplied by more than one system. It was felt that no ill effects would be experienced in zone number 5, the junction point of systems numbers 1 and 2. The layout of the conditioned space shows a large number of private offices, the occupants of which were away many times during the test period. At the beginning of the study three of the 16 zones were supplied with a dry-bulb thermometer and relative humidity recorder. In the other 13 zones observations were made of the wet- and dry-bulb temperatures by a sling psychrometer at three times each day as given:

8:05 to 8:50 A.M.—Entering the office.
12:30 to 1:15 P.M.—Re-entering the office.
3:40 to 4:25 P.M.—Leaving the office.

From the readings so obtained it was possible to get an idea of the trends in the various zones. It was seldom that any marked change occurred in these readings. When the remaining 13 recorders were received one was placed in each zone and the practice of daily observations discontinued except for the purpose of checking the calibration of each instrument.

The recorders were placed in such a position in each zone so as to be free from the effect of solar radiation, radiation from occupants, drafts, and other local interference. The approximate location of each recorder is shown in Fig. 1 by the letter *R*. The instruments were mounted on portable stands with the elements approximately 4 ft above the floor level. The glass cover of each instrument was painted to make the reading invisible to the occupants.

METHOD OF RECORDING DATA

Questionnaire cards of the type shown previously were distributed daily to each office employee, and the card for the preceding day was collected at the same time. Over 22,000 cards were collected in the study. The data from these cards were tabulated on ledger sheets to give a complete record of each individual's comfort reactions during the period of the study.

Question No. 5 of the questionnaire deals with the worker's reactions to the inside conditioned air after a short period following entrance, and it is therefore the only question concerning a person's reaction to the cooled space unaffected by entrance and leaving. This paper deals solely with this subject.

Questions 1, 2, 3, 4, 6 and 7 are concerned with the worker's reaction to the outside, the shock experienced by him upon entering the cooled space in the morning and at noon, the reaction experienced by him upon reentering the outside at noon and after work, and his continued reaction to the outside during the off work hours. These questions will be made the subject matter of another paper being prepared for publication in the near future.

After the workers had gained experience in judging their feelings and reactions to air conditioning they were given a special questionnaire and requested to discuss more fully their general reactions to working in an air conditioned office and the general effect of this experience on their daily life

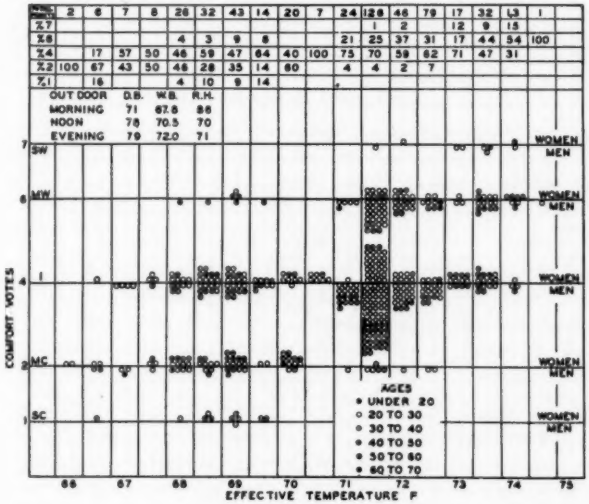

FIG. 2. SAMPLE OF DAILY PLOTTED DATA FOR AUGUST 31 SHOWING RELATION BETWEEN THE EFFECTIVE TEMPERATURE MAINTAINED AND (I) 4 IDEALLY COMFORTABLE, (II) 2 MILDLY COOL, (III) 6 MILDLY WARM, (IV) 1 SEVERELY COOL AND (V) 7 SEVERELY WARM

as compared to their friends and associates who did not enjoy this privilege. The results of this special study are most interesting, and will be made the subject of a third report.

In obtaining the data, particularly that collected under question 5, it was at first thought desirable to avoid doing anything to bring to the worker's attention any undue consideration of his comfort reactions. It was at first assumed therefore that they were comfortable whenever they did not register some other feeling. Most desirable air conditions were occasionally upset until they were definitely uncomfortable, in an effort to secure the limits in temperature and humidity which were comfortable to people as a whole. This method of securing data appeared to have the advantage of not causing

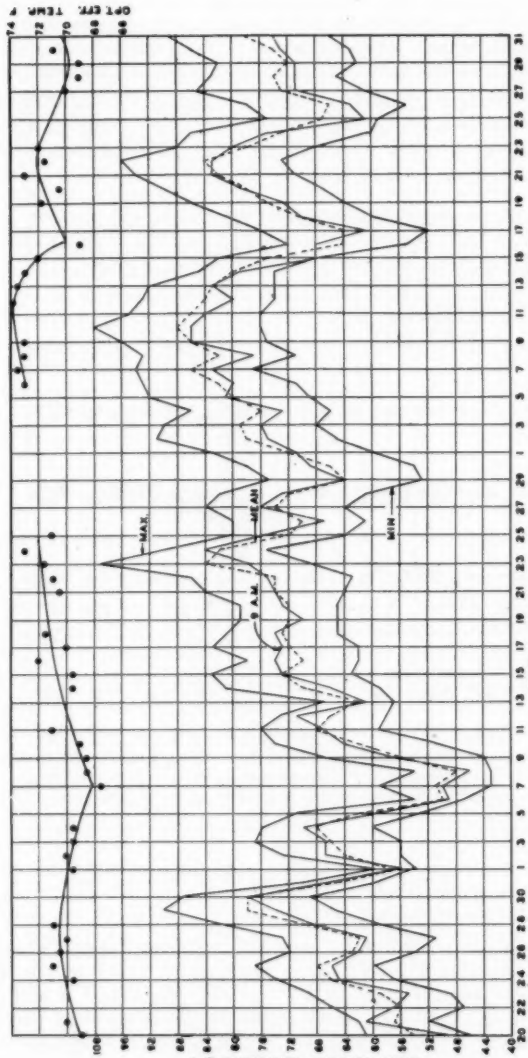

FIG. 3. DAILY OUTDOOR TEMPERATURE LOG AND OPTIMUM EFFECTIVE TEMPERATURE FOR INDOOR AIR CONDITIONING. MAY 20 TO JULY 31, 1937

a person to become conscious of the surrounding conditions until he was uncomfortable. After proceeding for approximately two months on this basis, it was felt that possibly more intelligent answers to question No. 5 could be had by asking the subjects to fill in their feelings at definite times. With this thought in mind, the person distributing and collecting the cards was instructed to circulate through the offices twice during the day and ask some workers to indicate their feelings. Approximately 35 different people each day were asked to supply this information. This procedure was followed until the completion of the study at the end of September.

It was also thought that by varying conditions in the three systems, it would be possible to obtain reactions over a wider range of air conditions. Since each system is under complete automatic control, it was readily possible to maintain quite widely different conditions with gratifying results.

ANALYSIS OF TEST RESULTS

A scale of feeling of warmth in which 4 represents ideal comfort, 3 and 5 comfortably cool and comfortably warm, respectively, 2 and 6 cool and warm, respectively, and 1 and 7 very cool and hot, respectively, has been used by the Research Laboratory over a period of years. In using this scale, it has been found that with untrained observers 3 and 5 are confusing, and they were eliminated from this study for that reason. In clarifying the meaning of the other numerical values in the scale to the workers, 4 was said to be ideally comfortable, 2 and 6 were called mildly cool or warm, and 1 and 7 severely cool and hot, respectively. The numerical values of 1, 2, 6 and 7 used in this study are therefore as nearly as can be evaluated the same as the same numerical values used in earlier studies. The numerical value of 4 used in this study may be said to be as nearly as possible equivalent to the three scale values of 3, 4 and 5 used in earlier studies. It is worthy of note that difficulty is always experienced in getting different persons to apply the same interpretation to their feelings. It may be assumed that the same interpretation is placed upon 4, or ideally comfortable, but that two different people may place different interpretations on mildly or severely cool.

A typical set of data collected on August 31 is plotted in Fig. 2. Each point represents the feeling expressed as 1, 2, 4, 6 and 7, discussed above, and plotted against the effective temperature in the particular zone in which the occupant was located at some time during this day. The many points for a given feeling at a certain effective temperature are grouped around a given ordinate line. The number of points does not make it possible to group them all on a given horizontal line. The group is indicated, however, as applying to this line, without any attempt to express fractional degrees of feeling. The expressions of comfort of the men are indicated below any ordinate line, while the expressions of women are indicated above, and the different age groups are indicated by the symbols noted.

Inspection of the entire plot, including all points for all age groups and for both sexes, indicates a maximum expression of ideal comfort, or 4, at 70.5 deg effective temperature, with larger percentages indicating 6 or 7 for higher temperature, and a greater feeling of coolness for lower temperature. The effective temperature for maximum comfort may also be

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determined by reference to the tabulated data, giving a percentage indicating various degrees of comfort at the top of the chart. This will indicate optimum comfort at approximately 70.5 deg effective temperature.

A statistical analysis of the same data for August 31 indicates that with

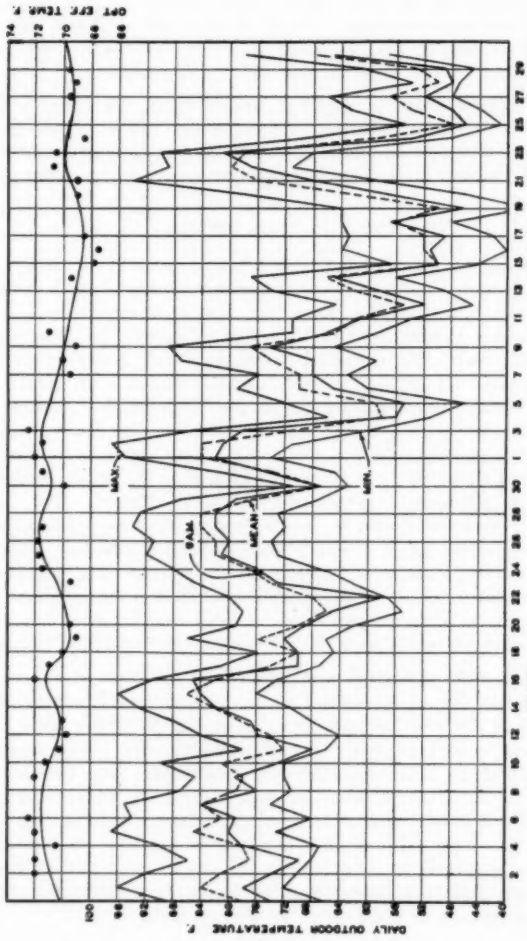


FIG. 4. DAILY OUTDOOR TEMPERATURE LOG AND OPTIMUM EFFECTIVE TEMPERATURE FOR INDOOR AIR CONDITIONING. AUGUST 1 TO SEPTEMBER 30, 1937

perfect sampling optimum comfort should be had at 71.5 deg effective temperature. The variation in the effective temperature for optimum comfort with different methods of analysis is quite typical of all the data analyzed. While the variation is of academic interest, it is not of any great significance

on most days. Undoubtedly, the statistical analysis gives indications nearer the true values, and was used on most of the data.

The data collected on each day on which observations were made between May 20 and September 30 are plotted in the upper part of the chart in Figs. 3 and 4, parallel with the log of the normal, minimum, maximum, mean and 9 o'clock daily temperatures for the same period. For the data collected up to and including July 31, the several zones conditioned by the different systems were all operated with a view of giving maximum comfort without reference to the study. Naturally during the days when these reactions were plotted, as in Fig. 2, they did not spread out over as many degrees of effective temperature from right to left, and as a result, the effective temperature for optimum comfort could not be determined with the same degree of certainty. For the period August 1 to September 30, occupants in the different zones, or at different times within the same zone, were given a range of effective temperatures designed to give a greater spread when plotted as in Fig. 2. The variations in effective temperature for optimum comfort for this latter period in the summer are therefore somewhat more significant than for the earlier period.

Variation in optimum effective temperature for comfort with time and therefore weather conditions are of great interest. The data for May and that for the latter half of September indicate a positive desire for a somewhat lower effective temperature in indoor air conditioned space. A range of from 69 to 71 deg effective temperature for May is indicated, while for the most of July and August an effective temperature range of from 69 to 74 deg is indicated. Daily variations are not very great, but a close inspection indicates some variations with changing weather, thus indicating a desire for a little lower effective temperature indoors during a cold spell, as from June 5 to 9, for the latter part of September, and other short periods, when indoor effective temperatures as low as 67 deg were desired. Likewise, high indoor effective temperatures were desired on or after very hot days on June 23, July 10 and July 22. It is significant that the highest indoor condition was at times desired on the day following maximum outside weather conditions. This variation is, however, not as great as many authorities have suggested, and does not follow closely variations in weather conditions. A maximum variation during July and August, which may be ascribed to variations from cool to hot spells, would seem to be about 5 deg, or from 69 to 74 deg effective temperature.

The data for any one day do not permit a very satisfactory determination of any variation of a desired optimum effective temperature for different age or sex groups, or for variation in indoor effective temperature with outdoor conditions. These variations are better shown for groups of data covering different periods, as in Figs. 5 to 7. In Fig. 5 are plotted all the observations of comfort for a period from August 2 to September 3 as percentages of those feeling comfortable at the various effective temperatures. This chart is based upon approximately 700 votes for any one effective temperature. With this great mass of data the variation in percentage of comfort for any effective temperature maintained is quite definite.

These curves indicate that men show a maximum of about 80 per cent of comfort at 70 deg effective temperature, while women show a maximum of about 70 per cent comfortable at approximately 70.6 deg effective temperature,

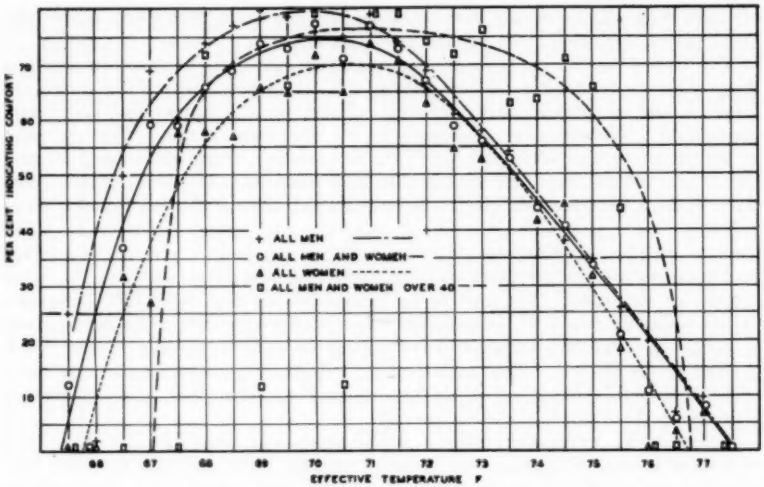
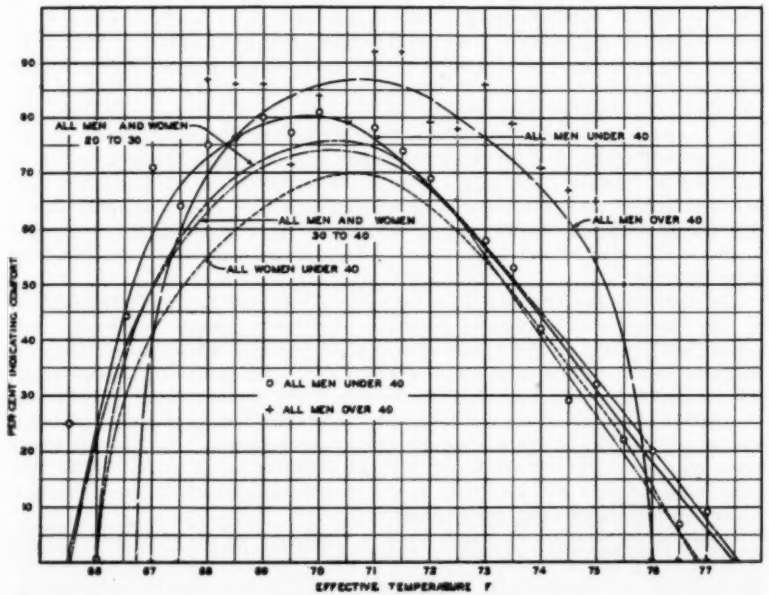


FIG. 5. RELATION BETWEEN EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR PERIOD AUGUST 2 TO SEPTEMBER 3, 1937. CURVE FOR ALL MEN AND WOMEN UNDER 40 COINCIDES WITH CURVE FOR ALL MEN AND WOMEN


ALL MEN AND WOMEN 20 TO 30
ALL MEN UNDER 40
ALL MEN AND WOMEN 30 TO 40
ALL MEN OVER 40
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PER CENT INDICATING COMFORT
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FIG. 6. RELATION BETWEEN EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR AGE AND SEX GROUPS. PERIOD: AUGUST 2 TO SEPTEMBER 3, 1937

while the entire group indicates optimum comfort at about half-way between these two extremes. This variation between men and women of all age groups is typical and is the same for all groups analyzed.

The optimum effective temperature for comfort for different age groups could not be determined with the same certainty as that for all men and all women plotted in Fig. 5. However, these relationships could be determined for a somewhat lesser degree of accuracy, and they are plotted in Fig. 6.

In order to show the variations in effective temperature desired in the late spring, mid-summer, and early fall, the percentage showing maximum comfort for different effective temperatures for periods from May 20 to June 10, for

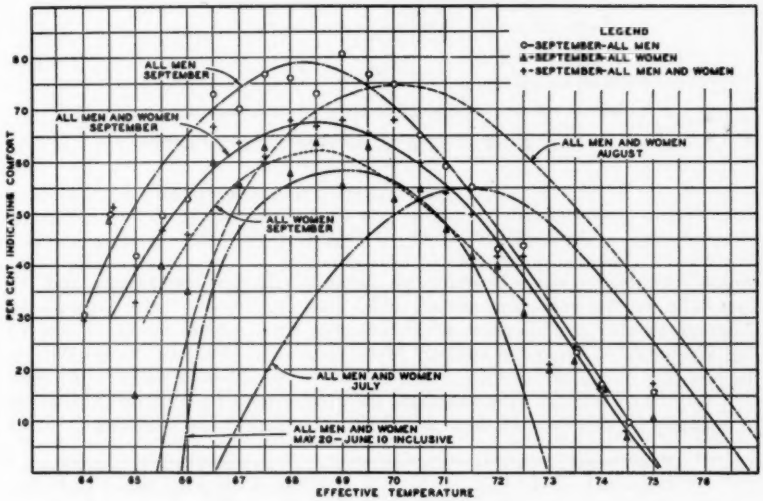

FIG. 7. RELATION BETWEEN EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR VARIOUS SEX GROUPS AND PERIODS. MAY 20 TO SEPTEMBER 30, 1937

July, August, and for September, are plotted in Fig. 7. It will be noted in reference to the weather log in Figs. 3 and 4 that these are periods of somewhat different average weather conditions. Optimum effective temperatures ranging from about 69 to 68 are indicated for May and September, and 71.5 and 70 deg for July and August, respectively. Inspection of the plotted data at the top of Figs. 3 and 4, however, indicates that on hot days optimum comfort may be had at as high as 74 deg effective temperature.

The variation in the effective temperature for optimum comfort plotted at the top of Figs. 3 and 4 indicates some relationship with weather conditions. In order to bring out this relationship, the optimum effective temperature for different days is plotted against minimum and maximum daily temperatures in Fig. 8, daily mean temperatures in Fig. 9, and the average mean temperature for a given day and the two previous days in Fig. 10. These three charts show

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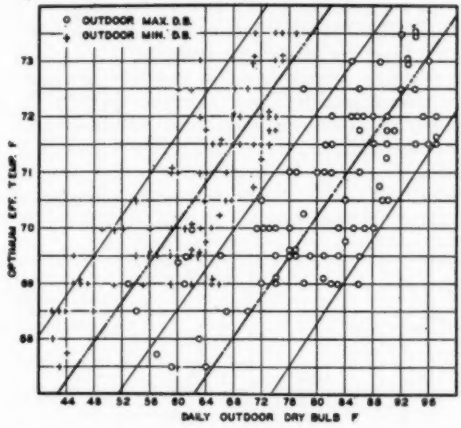

FIG. 8. RELATION BETWEEN DAILY OUTDOOR MAXIMUM AND MINIMUM DRY-BULB TEMPERATURE AND OPTIMUM EFFECTIVE TEMPERATURE

that a relationship exists between the optimum effective temperature and the four daily weather conditions against which they are plotted. Table 1 shows the different weather conditions at which effective temperatures of 68, 70 and 73 deg are indicated from the medium curve from each plot, and the number of degrees plus or minus tolerance which appears to be allowable in each instance.

It may be of interest to demonstrate the application of the statistical method of analysis of this type of data as used in this paper. In statistics a set of

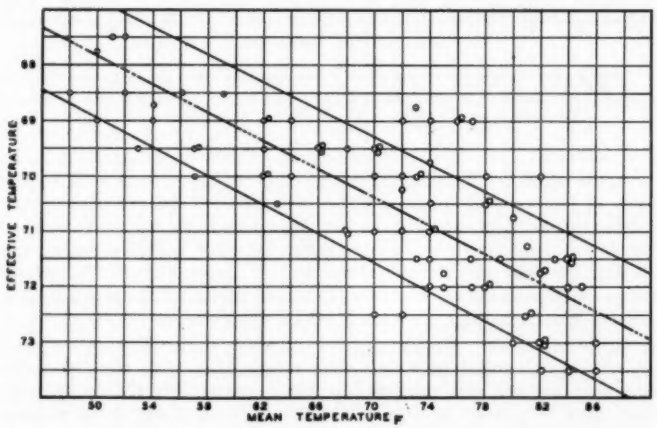

FIG. 9. RELATION BETWEEN DAILY MEAN DRY-BULB TEMPERATURE AND OPTIMUM EFFECTIVE TEMPERATURE

observations is considered a sample drawn from a parent population of possibly infinite size. If this sample is not too small and is selected in a manner that insures randomness, the probability that it is representative is good and can be calculated.

When individual measurements are repeated in a set of observations the result is a frequency. A set of comfort votes over a range of effective temperatures is an example of a frequency table, with the number of votes for each effective temperature a single frequency. In such cases it will be observed that a representative frequency curve of any particular sample takes a mound shape, often termed a *cocked hat*. It follows, therefore, that the comfort data approximates the normal curve:—

$$y = Ce^{-h^2x^2} \tag{1}$$

where:

e = the base of the natural or Naperian logarithms (2.71828)

C = a constant which determines the height of the curve, and

h = a constant which determines the spread.

This normal curve, because it is symmetrical, furnishes a basis for the determination of the accurateness of fit of any mound-shaped curve, whether it be badly skewed to either side, flattened out or peaked at the top. For these reasons it is also possible to calculate, with a definite probability, the expected frequencies of the parent population from the frequencies of the observed sample.

The maximum frequency is known as the *mode*. It should not be confused with the mean frequency (arithmetic mean). The mode for any comfort frequency chart is that particular frequency which is greatest, while the arithmetic mean frequency is the sum of the frequencies divided by the number of observations. There are a number of methods by which the mode for each comfort sample can be determined. An approximation can be obtained using the fact that the second degree equation:

$$ax^2 + bx + c = 0 \tag{2}$$

is a maximum, *when*

$$x = -\frac{b}{2a} \tag{3}$$

This approximation can be checked by calculating the expected frequencies of the parent population and has a special value when, as with a number of samples in the data, it is particularly difficult to determine the true mode. In this connection, it is desirable to point out that no mechanical process, such as statistics, can take into account all of the facts. In the case under consideration, after the mode was determined, the data for each sample were examined carefully. A few instances were found where a number of votes fell on the cold side of the comfort zone. It was evident, then, that the approximate mode calculated was not the true mode and adjustment was accordingly made.

A further check was applied in the determination of the true mode. This

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check also furnished the expected frequencies of the parent population. Because many of the samples involved a small number of observations, the Poisson Exponential Series was used for this purpose. It consists of the expansion:

$$e^{-m} + m\,e^{-m} + \frac{m^2}{|2}\,e^{-m} + \frac{m^3}{|3}\,e^{-m} + \qquad (4)$$

in which each term furnishes a frequency. Here e is the base of the natural logarithms and m is an exponent so chosen that the maximum expected frequency will occur for the effective temperature of the calculated mode. When

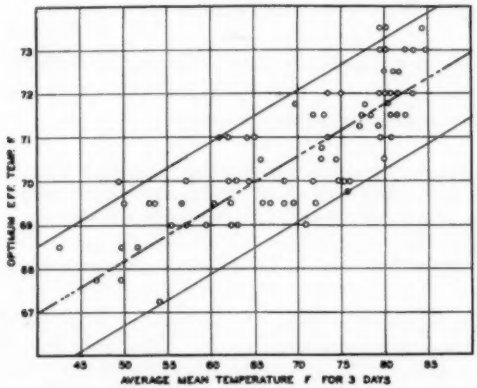

FIG. 10. RELATION BETWEEN THE AVERAGE MEAN DRY-BULB TEMPERATURE FOR A GIVEN DAY AND TWO PREVIOUS DAYS AND OPTIMUM EFFECTIVE TEMPERATURE

the expected frequencies are plotted it is possible to check the mode previously obtained. For an example, take the total ideal comfort data on both men and women for a period in August. These observed effective temperatures and votes are shown in Table 2. The total number of observations are 26 but, by inspection, it is evident that at least three are vacant. The temperatures advance in steps of ½ deg. Therefore, the mean (m) can be taken as 10.5. Inserting this value for m in the first term of the Poisson equation gives a value of 0.00001 for e^{-m}. Inserting the values for m and e^{-m} in the Poisson equation gives for the successive terms the values shown in the table, under the heading *Exponent Values*. The sum of the exponent values (0.27979) divided into the sum of the observed votes gives the factor 20683. When this factor is multiplied by each of the exponent values the expected votes for the parent population, shown in Table 2, are obtained. These votes have a sum equal to that for the observed votes in order that the accurateness of fit, at any point on the curve for ideal comfort (see Curve *A* in Fig. 11), can readily be observed. In the same manner, curves *B* and *C*, representative of cool and warm conditions,

TABLE 1—OUTSIDE WEATHER CONDITIONS REQUIRING INDOOR EFFECTIVE TEMPERATURES OF 68, 70 AND 73 DEG

WEATHER CONDITION	OUTSIDE TEMPERATURE FOR INDOOR REQUIREMENTS OF:			PLUS OR MINUS TOLERANCE FOR INDOOR EFFECTIVE TEMPERATURES
	68 Deg	70 Deg	73 Deg	
Minimum dry-bulb temp.—F	48	59	77	1.5 F
Maximum dry-bulb temp.—F	68	79	95	2.0 F
Daily mean dry-bulb temp.—F	51	67	91	1.2 F
Three day average, including that considered and the two previous days	49	66	91	1.5 F

TABLE 2—TABULATIONS FOR POISSON SERIES CALCULATIONS

EFFECTIVE TEMPERATURE	OBSERVED VOTES	EXPONENT VALUES	EXPECTED VOTES
65.5	1	....	0
66.0	0	....	0
66.5	13	0.00011	3
67.0	70	0.00050	11
67.5	38	0.00170	34
68.0	164	0.00470	91
68.5	264	0.00890	167
69.0	306	0.01550	286
69.5	322	0.02350	428
70.0	549	0.03150	570
70.5	292	0.03690	685
71.0	707	0.03950	744
71.5	608	0.03710	689
72.0	676	0.03200	591
72.5	453	0.02500	472
73.0	557	0.01900	355
73.5	226	0.01350	250
74.0	281	0.00850	167
74.5	133	0.00550	106
75.0	92	0.00325	63
75.5	21	0.00190	37
76.0	6	0.00090	20
76.5	3	0.00050	11
77.0	4	0.00023	5
77.5	0	0.00010	2
78.0	1	....	0
....	5,787	0.27979	5,787

are also obtained. With the total expected votes for each effective temperature obtained from the calculations for curves *A*, *B* and *C*, it is now possible to figure the percentage expectancy in votes for ideal comfort conditions as shown in curve *D*.

DISCUSSION OF RESULTS

The question of effective temperature requirements for air conditioning for different periods of the year has been the subject of intensive study since 1920. The early part of this work clearly demonstrated about 66 deg effective temperature as optimum for winter heating and air conditioning. More recent work on the requirements for summer cooling and air conditioning has indicated

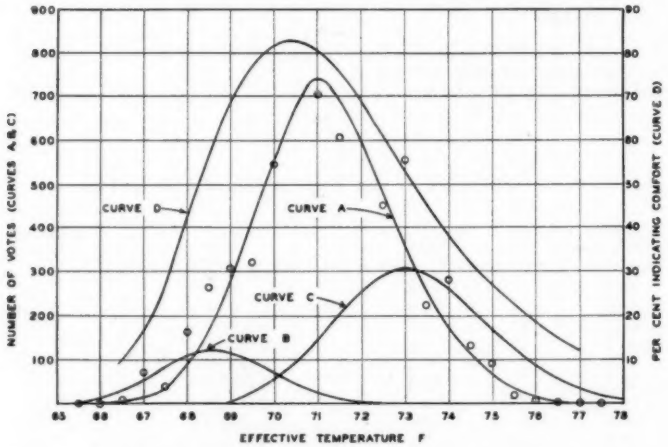

FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THEORETICAL NUMBER OF PEOPLE VOTING: COMFORTABLE, CURVE A; WARM, CURVE C; AND COOL, CURVE B. (POINTS ON CURVE A, ACTUAL OBSERVED). CURVE D SHOWS THEORETICAL PER CENT VOTING COMFORTABLE

from 68 to 71 deg effective temperature desired in the Toronto district, 69 to 74 deg effective temperature in Pittsburgh, and 71 to 74 deg effective temperature for College Station, Texas. These studies, made over the previous two summers, indicated a tentative conclusion for an optimum effective temperature for summer air conditioning of about 73 deg for Pittsburgh and any warmer climate, and about 71 deg for Toronto.

These studies gave no measurable indication of any variation in the daily requirement depending upon outside weather conditions. It was thought that the current study in Minneapolis, started in the spring and carried on until the fall, would clear up this question.

The daily requirements plotted in Figs. 3 and 4 should go a long way toward narrowing down the room for doubt and controversy. They do not, however, closely demonstrate a definite effective temperature as applying without ques-

tion throughout the cooling season. They do indicate some requirement for changing the indoor condition with violently changing outdoor weather conditions. The data for July and August show a maximum variation in desired indoor conditions from 69 to 74 deg effective temperature, or about the same range as concluded from last year's study for Pittsburgh. If, however, the indicated points in Figs. 3 and 4, for days when the outside mean temperature was 75 deg or lower, are eliminated, the total variation comes between 70 and 74 deg effective temperature. Since it is probable that in most cases heating would be required to maintain an indoor temperature above 69 or 70 deg when the outdoor mean was 75 deg or below, the provision for lowering indoor temperatures on these cool days should automatically take care of itself.

Experience gained at the Research Laboratory in its many studies indicates that a trained person is sensitive to a temperature variation of plus or minus 1 deg, but it is doubtful if a person not trained for such observation is sensitive to much less than plus or minus 2 deg; or at least, it may be said that a variation of plus or minus 2 deg is not a very important factor in a person's comfort. This reasoning would indicate that a continuous indoor condition giving 72 deg effective temperature in Minneapolis during the summer cooling season should give satisfaction, except during extremely cool spells, when the indoor temperature would automatically drop low enough to take care of itself, if artificial heating was not provided.

Hence, this study would seem to indicate the desirability of maintaining about 72 deg effective temperature throughout the cooling season in Minneapolis, with a drop to about 70 deg during May and June, and the latter half of September. Such a recommendation would differ by about 1 deg from the conclusions drawn from the earlier Laboratory studies [3] for Pittsburgh and Texas, which indicated a continuous indoor temperature of 73 deg throughout the cooling season. As pointed out in the earlier studies, the choice of a high indoor requirement within the allowable range of effective temperatures should serve both toward increased economy in cooling and to reduce the shock effect upon entering the cooled space. While such a recommendation would apparently give a condition slightly too high on some days when cooling is required in Minneapolis, it is doubtful if it would result in much discomfort for that reason, and therefore would serve toward increased efficiency in cost of operation and tend to decrease the cold shock effect.

ACKNOWLEDGMENT

The authors wish to acknowledge the guidance and assistance rendered by M. C. W. Tomlinson in helping to analyze statistically the great mass of data, which have added greatly to the value of the paper.

The Research Technical Advisory Committee on Comfort Air Conditioning is indebted to the Minneapolis-Honeywell Regulator Co. for their cooperation in providing the facilities of their general office building and the necessary personnel to observe and collect the data for these investigations, and to the Brown Instrument Co. for furnishing the recording instruments used in these studies.

[3] Loc. Cit. See Notes 1 and 2.

DISCUSSION

J. R. PARSONS:[4] There are just a few questions I wanted to ask at this time that are very close to my heart. I apologize if I bring the matter out at this time but I think it would be interesting.

In the first place, as Mr. Houghten says, the practical application is the thing that we have been working on for the last four years. Now, the first thing is the outdoor dry-bulb temperature. There is a very interesting amount of data that could be collected with reference to the outside effective temperatures. Just as a rather amusing thing, did you know that the sale of beer has a definite effective temperature, depending upon humidity at a definite dry-bulb? In summer, with a certain relative humidity, the sales are very large. With the same dry-bulb and a rise in humidity the sales fall off, although the effective temperature is going up. People change to water as a more cooling influence, but when you think that the breweries in their traffic departments have plotted this, it is an interesting correlation between effective temperature and the reaction of the human being, even to outdoor conditions.

Another point that is very close to me is this article that has been written which I have shown to many practical people, who have been interested, to prove certain points. When you are talking of effective temperature with the ordinary operating engineer, or the layman, or a board of directors, and you say you have found 71 in September or August as the optimum for this operation, you have to translate it as you would from Centigrade to Fahrenheit into what they mean as a 77 dry-bulb with something like a 50 relative humidity.

As I understand it, most applications in refrigeration and dehumidification in summer have a basis of about 50 per cent to which they guaranteed to drop the humidity. Here is a 70 dry-bulb and 50 relative humidity, moderate air motion or practically still air, and you say that is a 66 effective. Well, what better condition would you want, a perfect June day, a perfect September day, and you call that something else again; why not call it a 77 ET based on 50 per cent relative humidity?

One more question: do you not think it would clarify the subject if the index value was placed on a 50 per cent basis in which zone most of our operations are effected? We find then that you can demonstrate very readily what is out of line when the humidity goes up to 95 in August with 82 dry-bulb, as it did in New York for three days this past August. We calculate it to be an 89 effective. Everybody was affected by it and they went around with their coats over their arms and their collars off and all that sort of thing. A few days afterward we had 82 dry-bulb with 40 relative humidity, a perfectly delightful summer day, and the paper called attention to the fact that there was no change in the dry-bulb; it was not the heat, it was the humidity that had affected us.

[4] Consulting Physicist, New York, N. Y.

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SEMI-ANNUAL MEETING, 1938

AT the Semi-Annual Meeting 1938 held in The Homestead, Hot Springs, Va., June 20-22, over 200 members and guests of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS devoted their attention to a discussion of 13 technical papers during the four sessions. Because of threatening weather two sessions were held on Monday and single sessions on Tuesday and Wednesday.

Pres. E. Holt Gurney, Toronto, declared the Semi-Annual Meeting 1938 in session and introduced J. A. Donnelly, Largent, W. Va., General Chairman of Arrangements, who welcomed the visiting members and guests to Virginia.

President Gurney announced that weather conditions made it impossible to have the scheduled outdoor sports program and if it was the sense of the meeting that an afternoon session be held, three papers would be presented.

It was regularly moved, seconded and voted that an afternoon session be held at 2:00 p.m. and that three technical papers be discussed.

President Gurney opened the meeting and stated that if anyone present desired to reopen the discussion of the three papers presented at the Monday afternoon session, the opportunity would be afforded to them. No comments were offered.

Status of Research Work

The Committee on Research held two lengthy sessions and considered the reports submitted by the 24 technical advisory committees. Contracts for cooperative work were renewed with the University of Minnesota to continue investigations on air filters; the University of Illinois Medical School to investigate the effect of atmospheric conditions on cardiac cases; and the University of Pittsburgh to observe the reactions of patients, doctors and attendants to various atmospheric conditions in an operating room.

To assist in launching a three-year investigative program on air distribution, new agreements were approved with the University of Wisconsin, Case School of Applied Science to study the behavior of air discharge from grilles and when funds are available at the University of Illinois.

Cooperative agreements were authorized for work at Princeton University to study problems of sound control and the University of Toronto, Canada, for investigating the structure of insulating materials.

A summary of some work now under way is outlined herewith:

SENSATIONS OF COMFORT: C. Tasker, *Chairman.*

Negotiations have progressed whereby definite arrangements have been made to conduct comfort studies this summer in the Federal Reserve Building, Washington,

D. C., the Public Service Building, San Antonio, Texas, and the Metropolitan Life Insurance Building, New York City.

TREATMENT OF DISEASE: Dr. T. L. Hazlett, *Chairman.*

Studies are being continued on the effect of atmospheric conditions on cardiac cases at the Medical School of the University of Illinois by Dr. R. W. Keeton. This program is outlined in detail in the Annual Report of the Committee on Research 1937.

A cooperative study has been developed with the University of Pittsburgh Medical School where regular observations are now being taken on the comfort of the patient, doctor and his attendants in the operating room at the Magee Hospital to determine their reactions to various atmospheric conditions. At the same time efforts are being made to assemble the necessary instruments and personnel to determine the anesthetic content of the air in different parts of the room, the bacteria content of the air as well as the desirable physiological facts.

AIR CONDITIONING IN INDUSTRY: A. E. Stacey, Jr., *Chairman.*

The present objectives of this committee deal with the air conditions prevalent in industrial applications. Steps will be taken immediately to investigate the physiological reactions of individuals to conditions along the effective temperature lines of 85 and 87 deg with men at rest and stripped to the undershirt. These studies will be conducted in the Research Laboratory psychrometric rooms in Pittsburgh. The program is further intended to determine the critical point of heat loss from the human body for lines B, C and E which are represented on Fig. 8, page 74 of THE GUIDE 1938.

AIR CLEANING AND ATMOSPHERIC IMPURITIES: H. C. Murphy, *Chairman.*

The committee is considering a revision to the Society's Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. It is expected that some method of comparing air filter efficiencies by the use of a light meter will be considered in these revisions.

It is contemplated that one phase of the program now being outlined by the committee personnel will include a study in establishing the reduction in bacterial content of air before and after cleaning with different methods of application. Further it is felt that some study of the characteristics of the bacteria involved is essential particularly in view of the more recently accepted demonstration of the fact that pathogenic bacteria may be air-borne.

RADIATION AND COMFORT WINTER AND SUMMER AND EFFECT OF VARYING HUMIDITY ON RADIANT HEATING AND COOLING: J. C. Fitts, *Chairman.*

As a result of a series of preliminary conferences and by the exchange of correspondence with the committee personnel a tentative scope of the work to be projected is as follows: (1) A study based on primary sense reactions of the relative comfort derived from rooms with normal window arrangements from convected heat and from heat supplied by both convection and direct radiation; (2) A study of the effect of radiation in governing the sense of warmth of a person in an occupied space, together with the effect of dry-bulb temperature, moisture content and movement of the air already studied by the Society and included in the Effective Temperature Index or the Comfort Chart; (3) A study of the physiological reactions of persons, including skin temperature, comfort and health measurements, to rooms heated with and without radiation; and (4) A study of the relative effect on comfort and physiological reactions by radiation of varying wave-length, and the effect of the relative humidity of the air within the room on the effectiveness of such radiation.

As an immediate objective the committee is considering the initiation of studies covering item 1 to be accomplished at the Research Laboratory.

WEATHER DESIGN CONDITIONS: T. H. Urdahl, *Chairman.*

The program as outlined by the Chairman of this committee has suggested the following tentative plan for action as a result of a series of conferences and exchange of correspondence between the committee personnel: (1) That a bibliography be prepared and circulated to the committee members, outlining the work already done

and the manner in which the various data were collected; (2) That the factors which were not considered in any of these studies be brought out as problems for future consideration; and (3) That some method be adopted whereby a recommended procedure of collecting weather data be established.

The chairman of this committee has reported that this program is now ready for execution and it is expected that the necessary data will be assembled and correlated at the Research Laboratory.

RADIATION WITH GRAVITY AIR CIRCULATION: M. K. Fahnestock, *Chairman.*

The research work of this committee conducted at the University of Illinois may be divided into two parts: (1) that which has or may be done in a warm wall test booth; and, (2) that which may be done in a room heating testing plant, where test conditions simulate actual service conditions.

Under part 1 an extensive study of the inlet and outlets of convector heater cabinets with reference to their effect upon the heat output or capacity of the units has just been completed. These data will be presented in a technical paper. Future work in the booth will probably include tests on lightweight direct cast-iron radiators.

Under part 2, the remodelling whereby the test plant was enlarged to provide a test room 15 ft x 18 ft x 8 ft 6 in. in place of the two former 9 ft x 11 ft x 9 ft test rooms is practically completed. The first tests will be to correlate the performance of conventional tubular steam radiators and convectors in the 15 ft x 18 ft test room with their previous performance in the 9 ft x 11 ft test rooms. Future work will include studies with conventional direct cast-iron radiators, lightweight direct cast-iron radiators, and convectors, both free standing in the room, and recessed under windows in exposed walls.

HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION: G. L. Tuve, *Chairman.*

Work is being resumed in checking the various methods for calculating dehumidifying coils. Special attention is being given to the determination of surface heat transfer coefficients which may in turn be combined to give the overall coefficient for any desired operating condition.

The entire program is directed toward the formulation of uniform practices, which can then be recommended as the basis for a standard Test Code for rating and calculating surface coils. Five subdivisions of the general problem are covered in the present plans for experimental studies and correlations: (1) variation of air-side surface coefficients of both dry and wet coils; (2) effect of water velocities on the water-side coefficient of both refrigerating and heating coils, connected in counterflow and in cross-flow; (3) improvement of methods for measuring refrigerant-side surface coefficients of direct-expansion coils; (4) effect of coil depth or number of rows of tubes, and performance of very deep coils, (6 to 10 rows); and (5) relation of heat transfer to air friction and air velocity.

COOLING LOAD IN SUMMER AIR CONDITIONING: J. H. Walker, *Chairman.*

A preliminary report covering the research investigations observed at the Research Laboratory has been distributed to the committee personnel giving fragmentary data on a few walls in which the time lag due to solar radiation has been noted. A complete report of the program which is contemplated for study by the Research Laboratory this summer has been submitted to interested individuals and if financial assistance is obtained it is contemplated that the work can start immediately.

SOLID FUELS: W. A. Danielson, *Chairman.*

The general problem of this committee deals with utilization of fuels with the ultimate objective to provide better service to the consumer. The scope of work thus far outlined seems to emphasize the importance of collecting facts from recognized authorities and further attempting to correlate the work of other agencies doing investigative work in this field.

One of the important problems to be considered indicates that in the solid fuel field there is a seeming lack of approved test methods and means whereby comparative standards in performance of equipment can be evaluated. The importance of

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mutual education from the consumer's viewpoint will be emphasized with regard to cleanliness, convenience and comfort studies. Utilization of solid fuels has been generally classified by the committee as dealing with (1) equipment, (2) method of firing, (3) drafts, and (4) fuel and ash handling.

SUMMER AIR CONDITIONING FOR RESIDENCES: M. K. Fahnestock, *Chairman.*

The research work of this committee conducted in the Research Residence has been in cooperation with the University of Illinois and the National Warm Air Heating and Air Conditioning Association. This study is to be continued during the summer of 1938 with the following objectives: (1) to obtain more information on the practicability of using a small capacity condensing unit in severe warm weather; (2) to obtain more information on the feasibility of cooling the first floor during the day and the second floor at night; (3) to obtain data in regard to the rate at which the temperature of the air in a space, where cooling has not been in progress, can be reduced with several different cooling capacities; and (4) to study a possible method of alleviating the unsatisfactory fluctuations in indoor relative humidity which occur in mild weather in a residence, where the dry-bulb temperature is maintained by starting and stopping the condensing unit with a dry-bulb temperature thermostat.

AIR DISTRIBUTION AND AIR FRICTION: J. H. Van Alsburg, *Chairman.*

A comprehensive program for investigations in this field has been outlined with the work to be accomplished at various cooperative institutions and the Research Laboratory.

At the University of Illinois it has been suggested that tests be conducted in their 15 x 18 ft cold room where the walls, ceilings or floors may be heated or cooled in order to determine the advantage of location of air outlet with specific regard to supply and return.

The program contemplated at the Case School of Applied Science involves a comparison of instruments used for air velocity measurement and the behavior of air discharge from grilles into large unconfined spaces.

At the University of Wisconsin a cooperative project is being considered which will consist of testing the effect of approaches to an air outlet and possibly further tests in large rooms with variable shaped openings.

HEAT REQUIREMENTS OF BUILDINGS: P. D. Close, *Chairman.*

As THE GUIDE is materially lacking in data pertaining to heat losses into the ground, it has been recommended that a study be conducted at the Research Laboratory to establish proper heat loss coefficients. Tests have been planned to determine the heat loss into the ground by use of a suitable heat flow meter. The heat loss through top floor ceilings and under pitched roofs has also been considered by this committee and it is expected that a revised method of calculating these losses will be submitted to the Guide Publication Committee.

AIR CONDITIONING REQUIREMENTS OF GLASS: M. L. Carr, *Chairman.*

The work of this committee is actively progressing as a result of the program established last year with substantial earmarked funds to conduct current investigations. A technical paper reporting the work thus far accomplished by this group was presented at the Semi-Annual Meeting of the Society entitled Heat Transfer Through Single and Double Glazing.

INSULATION: W. A. Danielson, *Chairman.*

A comprehensive program of research activity was developed by this committee a year ago with particular interest displayed in the development of codes for testing conductivity and heat transfer. Some of the subjects dealing with the fundamental aspects of insulation constitute the general work of the committee and may be enumerated as follows: (1) the effect of regain moisture on the conductivity of insulating materials, (2) the effect of condensed moisture on the conductivity of insulating materials in a structure, (3) effect of the thickness of the insulating

material accepted for test and the conductivity value obtained, (4) effect of condensation on the surface of reflective insulation, (5) effect of fiber size and fiber arrangement within an insulating board on its conductivity, (6) relation between the heat capacity of an insulating material and the rate of heat penetration through a structure in which it is installed, and (7) the degree of settling found for different fill materials after they are placed in a structure.

SOUND CONTROL: J. S. Parkinson, *Chairman.*

It is the consensus of this committee that a code is both necessary and desirable for testing and rating the noise of air conditioning equipment. A suggested code has been outlined and submitted to the members of this committee for detailed consideration. A request has also been forwarded to various air conditioning manufacturers soliciting their cooperation in furnishing the committee with any available data now being used in predicting the noise levels in a room and applying this information to the selection of equipment and application of sound absorptive materials.

COOLING TOWERS, EVAPORATIVE CONDENSERS AND SPRAY PONDS: B. M. Woods, *Chairman.*

Primary factors affecting this type of equipment have been under consideration by this committee. In connection with cooling tower performance the problems for study have been generally classified as those dealing with the gross factors usually considered for a given tower, such as wind velocity, wet-bulb temperature of air striking the tower, temperature of water on the tower, and loading of water on a tower in gallons per minute per square foot of area.

The cooling effect of various thicknesses of water maintained on roof decks is being studied by this Committee and during the current summer it is expected that various observations will be made on test cubicles to ascertain actual surface temperatures.

CORROSION IN STEAM SYSTEMS: A. R. Mumford, *Chairman.*

This committee is actively engaged in developing a program to determine the probable effect of the rate of condensation upon the oxygen content of a condensate leaving a radiator. This work is a continuation of that outlined in the Annual Report of the Committee on Research 1937.

CORROSION IN AIR CONDITIONING EQUIPMENT: A. E. Stacey, Jr., *Chairman.*

At the annual meeting of the Society in New York last January a memorandum was reviewed covering the work thus far reported by this committee. Since that time additional correspondence has been exchanged between the committee members and it is expected that a report will soon be released dealing with this subject.

On Wednesday morning President Gurney called for the Report of the Tellers on the ballots for the A. S. H. V. E. Standard Code for Testing Stoker-Fired Steam-Heating Boilers. The report showed 348 legal ballots, of which 345 voted approval of the Code and three disapproved.

Amendments to Research Regulations

The amended Regulations Governing the Committee on Research were presented and on motion of Mr. Fleisher, seconded by Mr. Tasker, it was unanimously voted to approved the following articles:

ARTICLE II, *Organization—Section 2. Chairman and Vice-Chairman, Committee on Research*—A Chairman and a Vice-Chairman of the Committee on Research shall be appointed each year by the President of the Society immediately after election, such appointments to be made from the membership of the Committee on

Research. The member appointed Chairman shall have served on the Committee not less than one (1) year.

OMIT SECTION 3. Section 3. Vice-Chairman—A Vice-Chairman shall also be elected in the same manner and for the same period as stated for the Chairman in *Section 2.*

Section 4. Research Executive Committee—There shall be a Research Executive Committee consisting of the Chairman and the Vice-Chairman of the Committee on Research and three (3) other members of the Committee on Research, appointed by the Chairman immediately following the Annual Meeting.

ARTICLE III, *Duties of Officers—Section 1. Chairman*—The Chairman of the Committee on Research shall preside at all meetings of the Committee and the Research Executive Committee. He shall be an ex-officio member of the Council. He shall approve all bills against Research activities before payment is made.

Section 3. Director of Research Laboratory—The Director of the Research Laboratory shall be in direct charge of the Laboratory under the direction of the Chairman of the Committee on Research. He shall be responsible to the Committee on Research, and shall cooperate with the Chairman of the Committee in every way possible. He shall approve all bills against Research activities before payment is made.

ARTICLE IV, *Duties of Committees—Section 1. Committee on Research*—The Committee on Research shall have general charge of all Research activities including the making of the necessary contracts for rental or purchase of equipment or materials. This Committee shall select and engage a Director of the Research Laboratory, subject to the approval of the Council and such assistants of the Research Laboratory and the Committee on Research as may be required, or the authority for the selection and employment of the Laboratory assistants may be delegated to the Director of the Laboratory by authority of the Committee if the Committee so desires. The Committee shall determine all salaries, approve all cooperative agreements and determine the order in which the subjects shall be investigated by the Research Laboratory. Any proposed expenditure of Research funds outside of the approved budget shall be approved by the Committee on Research before the expenditure is made.

ARTICLE V, *Government—Section 3 (a). Approval*—All acts of the Committee on Research other than those specifically authorized shall be subject to review by the Council.

Section 7 (a). Payment of Bills—All bills against Research activities shall be presented to the Director of the Laboratory and to the Chairman of the Committee on Research, who will present approved bills to the Secretary of the Society.

The Secretary shall check and, if correct, approve for payment all items when covered by the budget, or emergency expenditures approved in writing by the Chairman of the Committee on Research.

A summary of these accounts shall be made at least every thirty (30) days on a form approved by the Council and presented to the Treasurer of the Society, who shall draw a check against the Research Fund, payable to the American Society of Heating and Ventilating Engineers, for the total amount of the approved bills, which check shall be deposited in the Secretary's special account. The Secretary shall draw against this amount in settlement of all approved expenditures.

Section 7 (b). The Secretary shall have the authority to pay salaries, traveling expenses, and petty cash in accordance with the budget.

Section 8. Accounts—The book of accounts shall be kept at the headquarters of the Society, and shall be audited each year at the time the Society books are audited and by the same certified public accountant who examines the Society accounts. The accountant's report shall be presented at the Annual Meeting of the Society by the Chairman of the Committee on Research as the financial report on research activities for the year.

Amendments to Constitution

The Secretary reported the receipt of a petition signed by ten members for the amendment of Article C-V, *Section 1,* of the Society's Constitution. President Gurney asked that the original article and the amendment be read by the Secretary.

ARTICLE C-V, *The Council:*

Section 1. The affairs of the Society shall be managed by a Board of Directors chosen from among the persons entitled to vote, which shall be styled "The Council." It shall consist of the President, First Vice-President, Second Vice-President, Treasurer, the last Past President and twelve (12) elected members, four (4) of whom shall be elected each year to hold office for three (3) years. The Secretary may take part in the deliberation of the Council, but shall have no vote therein.

To be amended as follows:

Section 1. The affairs of the Society shall be managed by a Board of Directors chosen from among the persons entitled to vote, which shall be styled "The Council." It shall consist of the President, First Vice-President, Second Vice-President, Treasurer, the last Past President and twelve (12) elected members, four (4) of whom shall be elected each year to hold office for three (3) years. The Chairman of the Committee on Research shall be an ex-officio member of the Council. The Secretary may take part in the deliberation of the Council, but shall have no vote therein.

Resolutions

J. F. S. Collins, Jr., Pittsburgh, Pa., offered the following resolutions of appreciation:

Before closing this pleasant instructive meeting, we the members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS assembled in Hot Springs, Va., for the Semi-Annual Meeting 1938, wish to thank those who are responsible for making it an outstanding success, therefore,

BE IT RESOLVED, that we are indebted to the Committee on Arrangements who under the able leadership of their general chairman, J. A. Donnelly, and his committee chairmen, have worked so faithfully and efficiently.

THAT, we appreciate the messages brought to us by the very able speakers at our technical sessions and at our banquet to stimulate us through the coming months.

THAT, we enjoyed the scenic and inspection trips provided by the Arrangements Committee.

THAT, we are appreciative of the service and courtesies extended by the management of The Homestead.

THAT, we thank the press for its generous support in giving space and publicity to the activities and to the papers presented at our sessions.

BE IT FURTHER RESOLVED, that to all those who have contributed toward making this meeting a success, we wish to express our sincere appreciation.

On motion of Mr. Collins, seconded by Mr. Fleisher, it was unanimously voted to adopt these resolutions.

President Gurney asked if there was any unfinished business to be brought before the meeting or any new business and, in the absence of any further business, a resolution to adjourn was adopted.

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PROGRAM SEMI-ANNUAL MEETING, 1938

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
THE HOMESTEAD, HOT SPRINGS, VA.
JUNE 19-22, 1938

Sunday, June 19
(Eastern Standard Time)

10:00 A.M. Meeting Committee on Research—Pilgrim Room
2:00 P.M. Meeting of the Council—Pilgrim Room
2:30 P.M. Committee on Air Conditioning Requirements of Glass—Room 108

Monday, June 20

8:30 A.M. Registration
9:30 A.M. Committee on Solid Fuels—Room 108
9:30 A.M. First Session
Technical Papers:
A Test Method for Air Filters, by R. S. Dill
The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr., and M. H. Dean
The Condensation Nuclei Content of the Air as Related to Air Freshness, by R. A. Nielsen
Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter, by Frank B. Rowley and Richard C. Jordan
2:00 P.M. Committee on Air Distribution and Air Friction—Room 108
Committee on Radiation and Comfort Winter and Summer—Room 110
Committee on Weather Design Conditions—Room 112
8:00 P.M. Committee on Summer Air Conditioning for Residences—Room 108
Guide Publication Committee—Room 112

Tuesday, June 21

8:30 A.M. Committee on Sensations of Comfort—Main Dining Room
9:30 A.M. Committee on Air Cleaning and Atmospheric Impurities—Room 108
Second Session
Technical Papers:
Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller, Leighton Orr and Alan C. Byers.
Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson
Performance of Surface Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. J. Seigel
Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick
The Measurement of Heat Transfer by Free Convection from Cylindrical Bodies by the Schlieren Method, by L. M. K. Boelter
12:30 P.M. Nominating Committee—Pilgrim Room
2:00 P.M. Committee on Insulation—Room 108

Wednesday, June 22

9:30 A.M. Third Session
Amendments to Regulations Governing Committee on Research
Technical Papers:
Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker
Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke and W. H. Badgett

Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson
General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski

2:00 P.M. Committee on Heat Requirements of Buildings—Room 108

ENTERTAINMENT EVENTS

Monday, June 20

11:30 A.M. Concert for Ladies
2:00 P.M. Golf Tournament—Cascade Course (18-Hole Flag Tournament)
2:30 P.M. Tennis, Croquet, Horseshoe Pitching. (Sign up at Registration Desk)
Carriage Ride to Dunns Gap. (Secure tickets at Registration Desk)
7:00 P.M. Get-Together Dinner
8:00 P.M. Movies in Auditorium
8:30 P.M. Card Party
10:00 P.M. Dancing—Crystal Room

Tuesday, June 21

10:30 A.M. Ladies Golf and Putting Contest
2:00 P.M. Golf Tournament—Homestead Course (18-Hole Medal Play; Research and Chapter Cups)
Motor Trip and Inspection of Covington Paper Mill
2:30 P.M. Ladies Bridge and Tea
7:00 P.M. Semi-Annual Banquet and Dance. *Speaker*—Dr. C. M. Newman of Virginia Polytechnic Institute on Some Elements of American Humor

Wednesday, June 22

2:00 P.M. Golf—Optional on Homestead or Cascade Courses

COMMITTEE ON ARRANGEMENTS

J. A. Donnelly, *General Chairman*

Perry West, *Reception*
I. B. Helburn, *Registration*
E. W. Klein, *Finance*
J. K. Peebles, Jr., *Publicity*
J. A. Shanklin, *Banquet*
L. F. Nordine, *Golf*
R. McC. Johnston, *Entertainment and Sports*
Albert Buenger, *Ladies*

No. 1090

A.S.H.V.E. STANDARD CODE FOR TESTING STOKER-FIRED STEAM-HEATING BOILERS *

I. INTRODUCTION

1. The purpose of the Code for Testing Stoker-Fired Steam-Heating Boilers is to provide a standard method for conducting and reporting tests to determine heat efficiency and performance characteristics; this Code does not fix the conditions that must be fulfilled for the acceptance or approval of the combination of boiler and stoker. The Code recognizes that tests of stokers and boilers may be made under different conditions for different purposes, and with complete or limited facilities for conducting the tests.

It should be appreciated that in all cases to which this Code applies there are two devices, that is, boiler and stoker, involved in the test. Both devices influence the results obtained. Two different stokers placed successively in the same boiler may not produce identical results. To view the results as if they pertained only to the boiler would be an incorrect viewpoint.

2. Standard Form. Table 1, Standard Form, Solid Fuels—Stoker Fired is arranged for recording the full data for a continuous test called for by this Code.

3. Short-Test Form. Table 2, Short-Test Form, Solid Fuels, Stoker Fired is for intermittent tests or for continuous tests in which a complete heat balance is not desired. The following determinations are not required:

(a) Proximate analysis of fuel, except as required for record.
(b) Ultimate analysis of fuel.
(c) Complete flue gas analysis; CO_2 is required for record.

4. Continuous and Intermittent Tests. Two general types of tests are recognized; they will be designated as Continuous and Intermittent. A Continuous test may be defined as one where the stoker operates at a substantially constant rate throughout the duration of the test; it may use either the Standard or Short-Test Form. An Intermittent test is one where the stoker either shuts off or operates at a lower rate at intervals throughout the test; it is designed to cover the determination of only the stoker and boiler efficiency and may use only the Short-Test Form.

II. DEFINITIONS

5. Furnace Volume is the space which may be used for combustion. It shall be considered as that space enclosed by the hearth or grate surface and the side walls or water legs up to the crown sheet or equivalent. If a bridge wall is used, that

* Adopted 1938 by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., and the *Stoker Manufacturers Association*, Chicago, Ill.

COMMITTEE: C. E. Bronson, *Chairman;* L. A. Harding, H. M. Hart, F. C. Houghten, A. J. Johnson, J. F. McIntire, D. W. Nelson, Percy Nicholls, R. A. Sherman, and E. C. Webb.

space back of the bridge wall above the horizontal plane passed through the top of the bridge wall shall be included.

6. Hearth or Grate Area is the projected area bounded by the furnace walls at the hearth or grate level except for spillover stokers where it is the projected area of the retort.

7. Combustion Rate is the number of pounds of fuel, as fired, fed by the stoker per hour, less the fuel equivalent of the combustible in the ash. The fuel equivalent of the combustible in the ash shall be taken as the weight of the combustible multiplied by the ratio of 14,600 to the calorific value of the fuel as fired.

8. Heat Release Rate, in Btu per hour, is the product of the combustion rate and the calorific value of the fuel.

9. Boiler Output in Btu per hour, is the quantity of heat delivered at the boiler nozzle with the boiler normally insulated.

10. Efficiency of Stoker and Boiler is the percentage of the heat in the fuel fired which is delivered at the boiler nozzle when the boiler is normally insulated.

III. SET-UP OF STOKER AND BOILER TO BE TESTED

11. Stoker and Boiler when set up for the purpose of commercial testing shall be assembled according to the manufacturer's directions.

12. Setting of the stoker in relation to the boiler shall be as agreed to by all interested parties. NOTE: The report of a test shall clearly describe the manner of installation.

13. Boilers already installed shall be arranged to comply with the following conditions as nearly as possible, or as shall be agreed to by all interested parties.

14. Boiler Covering. The boiler should be covered with such heat insulation as is recommended or commercially furnished by the manufacturer; if tested uninsulated or partly insulated, an allowance shall be made for that portion of the heat lost from the bare metal steam surface which might have been saved by covering. The allowance shall be computed from the formula given with the tables of this Code. In no case shall allowance be made for uninsulated portions which would also be uninsulated in the commercial product.

15. Steam Outlet Piping. All steam outlet pipes from the boiler shall always be well covered, up to and including the steam separator.

16. The Vertical Outlets from the boiler leading to the separator shall be in number, arrangement, and size according to manufacturer's directions and shall not rise vertically from the boiler nozzle more than 30 in. before entering the steam separator.

17. The Steam Piping leading from the separator shall have a slope so that water condensed in it will drain away from the separator. The drain pipe from the steam separator shall include a U-leg sufficiently long to prevent the steam blowing out, and with a short inverted U-leg at the outlet end so that the drip may be caught in a container. Re-evaporation of water drained from the separator shall be prevented by use of a suitable cover on the container.

18. The Steam Valve on the boiler steam outlet piping shall be placed beyond the steam separator.

19. Feed Water Piping. The feed water pipe should connect to the boiler where the returns would normally come. The temperature of the feed water shall be read from a thermometer inserted into the feed line near the boiler but not so close that the temperature will be affected by heat conducted from the boiler. The thermometer may be in a well or cup, or it may be inserted directly into the feed water. The latter method is recommended to avoid errors introduced by heat conduction of a well.

A valve shall be placed on each side of the thermometer position. The valve between the thermometer and the boiler shall serve as a shut-off valve. The valve

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between the thermometer and feed water supply shall be used as a regulating valve only. It is recommended that this valve be an automatic one for maintaining a uniform water level in the boiler. If only one valve is used, it shall be placed between the thermometer and the boiler.

All boiler-water connections, including blow-off pipes, must be exposed to view, so that leakage may be observed and either stopped or measured.

20. Water Gage Marking. The water gage shall have a permanent mark to indicate the water level recommended by the manufacturer; if there is no permanent mark, for the purposes of these tests, a mark corresponding to the level recommended by the manufacturer shall be put on the water column gage glass.

21. Smokehood Connections. The boiler shall be connected with a short, direct smoke pipe to a source of draft. All joints shall be thoroughly sealed and maintained so during the tests.

The smokehood and the smoke flue shall be insulated with not less than 1 in. thickness of insulation; the smoke-flue insulation shall extend at least one equivalent flue diameter beyond the point where the flue-gas temperature is measured.

There shall be no check draft damper between the boiler and the point where the flue-gas sample is taken or the flue-gas temperature measured; if one is incorporated in the boiler, it shall be thoroughly sealed during all tests.

The choke damper, if incorporated in the boiler, shall be put in place, opened wide, and kept so during all tests. A damper shall be placed in the smoke pipe between the source of draft and the point where flue-gas measurements are made. This shall be called the regulating damper. A check damper, either manual or automatic, may be placed between this damper and the source of draft, if desired. An automatic damper is recommended. This check damper shall not be placed closer than two pipe diameters to the point where readings are being taken.

22. Source of Draft. The draft may be produced by a chimney or by a fan or other arrangement for providing induced draft.

23. Cleaning of Boiler. The water spaces of the boiler shall be thoroughly boiled out with a cleaning solution, then thoroughly rinsed with clean water.

24. Soot, Ashes, and Dust. The heating surface, furnace, and flues shall be clean and free from soot, ashes and dust at the beginning of the test.

IV. INSTRUMENTS AND MEASURING APPARATUS

25. Steam Output. The weight of water evaporated may be determined either by condensing the steam and weighing the condensate, designated as Method *A*, or by weighing the water fed to the boiler, designated as Method *B*.

Method A. Water fed to the boiler need not be weighed. The steam after passing through the separator shall be led to a condenser through a connecting steam pipe sloping toward the condenser. The condenser shall be so set up that all the condensate will drain quickly to the outlet. The condensate shall be caught in suitable tanks and weighed.

The condenser shall be of the closed type, preferably with outside joints, and of sufficient capacity that the temperature of the condensate shall not exceed 150 F. Before and after each test it shall be tested for freedom from leakage by putting it under full water pressure.

Method B. The weight of water fed to the boiler shall be determined in an accurate and suitable manner. The most convenient weighing arrangement depends on the size of the boiler, but it shall be such that the weight fed can be determined at any instant. The use of tanks fitted with calibrated floats or gage glasses is permissible.

The water may be fed to the boiler by gravity, air pressure, or feed pumps; any leakage between the measuring tanks and boiler shall be weighed and deducted from the water fed.

26. Weighing Scales. Accurate scales of suitable size shall be provided for weighing separator water, feed water, fuel, and all refuse removed from the furnace.

27. Draft and Pressures. A suitable number of draft gages shall be provided and so arranged as to determine the pressure difference between the room atmosphere

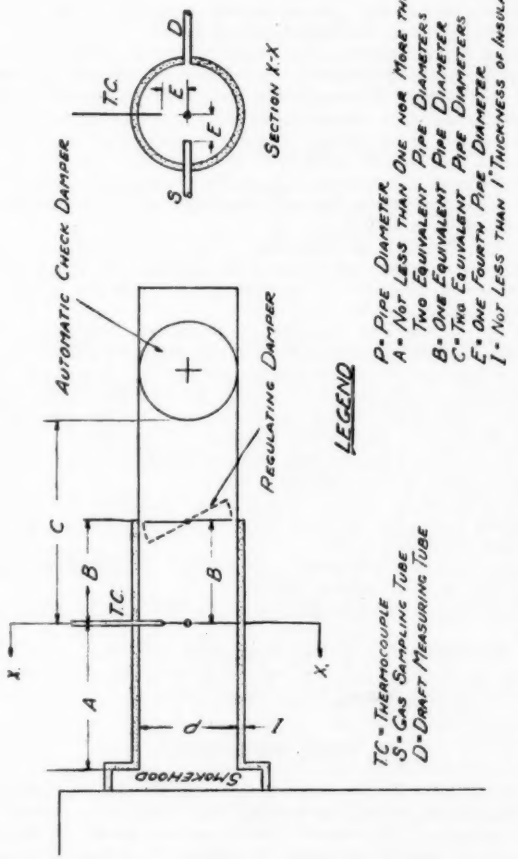

Fig. 1. Location of Thermocouple, Gas Sampling Tube, Draft Tube, and Dampers

and the stoker's windbox, the furnace, and the smokehood. Draft measurement shall be made with draft gages reading to 0.01 in. of water. Draft gages shall be checked for zero reading each hour.

28. The Smokehood Draft shall be measured at least one pipe diameter before the regulating damper. The connection into the flue shall have its end square and shall be set at right angles to the flow of the gases.

29. Furnace Draft. The connection for measurement of the furnace draft shall be made through a hole in the side or front of the furnace at a level approximately midway between top and bottom of the furnace door.

30. Stoker Windbox Pressure. The stoker windbox pressure shall be measured at a suitable point in the plenum chamber from which air is delivered to the grates or the tuyeres. The connection shall have its end square and flush with the inner wall of the plenum chamber and shall be so placed as to read the true static pressure.

31. Steam Separator. The steam separator shall have sufficient volume to catch slugs of water and to prevent their passing into the steam line; it shall also efficiently separate entrained water from the steam carrying it.

32. Temperature Measurements. Accurately calibrated instruments shall be provided for all temperature measurements. A mercury thermometer graduated to not more than 1 F is preferable for measuring feed water temperature. It is recommended that thermocouples, in preference to other types of thermometers, be used to measure the flue-gas temperatures. The temperature shall be measured in the smoke pipe not less than one nor more than two pipe diameters beyond the smoke-pipe collar. The thermo-junction or bulb of the thermometer shall be placed as shown in Fig. 1. The instrument used with the couple should be sensitive to at least 5 F and have a range up to 1200 F.

33. Sampling and Analysis of Flue Gas. The sample of flue gas for analysis shall be taken from the smoke pipe at the point at which the temperature is measured in such a manner as to give a fair average sample of the gas stream. An Orsat or equivalent gas analyzer shall be used.

An open end ⅛ in. iron pipe reaching one-fourth way across the flue pipe is usually satisfactory, but the probable mixing of the gases should be considered and the open end so placed as to be in the average gas stream. The use of a perforated pipe may be advisable in certain cases and shall be permissible. For gas temperatures of over 750 F the portion of the collecting pipe extending into the flue should be clay or silica, because the iron at this temperature may reduce CO_2 to CO. Air leakage around draft tubes, thermocouples, and gas-sampling tubes shall be prevented by using suitable packing or seal around the tubes.

As an average value for the test period is desired, the correct and simplest method is to collect samples of the flue gas at a constant rate into a bottle and to take the samples for analysis from those bottles.

If recording gas analyzing instruments are provided, they shall be checked every hour with the Orsat or other manual apparatus.

34. Smoke Readings. Smoke readings shall be made by the Ringelmann Chart Method.

35. Pressure Readings. A calibrated steam gage, a mercury column, or a mercury thermometer graduated to 1 F, and inserted directly into the steam space of the boiler shall be used for determining the steam pressure.

V. FUEL SAMPLING

36. (a) As the hopper of the stoker is filled during the test, samples shall be taken, in small increments, at regular intervals and stored in a covered water-tight container in a cool place. The sample shall be approximately 10 per cent of the fuel fired; the correct amount can readily be obtained by taking as a sample every tenth shovelful as the hopper is filled. In no event shall the total sample be less than 100 lb.

The gross sample shall be quickly reduced to a laboratory sample of 15 lb by the methods described in the current *Standard Method of Sampling Coal* of the *American Society for Testing Materials, Designation D21,* and placed in a sealed container for transportation to the laboratory.

(b) When a number of tests are to be conducted with the same lot of fuel, it

shall be permissible to sample the entire lot for ultimate and proximate analysis and calorific value as specified in the current *Standard Method of Sampling Coal* of the *American Society for Testing Materials, Designation D21.* When this method is used, samples shall be collected during each test, as in Paragraph 36*a* for determination of moisture and ash, and the ultimate and proximate analyses and calorific value for the gross lot corrected for same, as applied to the individual tests.

37. (a) After the ash and refuse has been weighed, it shall be reduced to a 15 lb laboratory sample for determination of its combustible content.

(b) In stokers equipped with automatic ash removal mechanism, this shall be disconnected and the ash collected directly from the ashpit when the test is less than 24 hours in duration.

In stokers operating on the principle of removal of ash as clinker, only the fused clinker shall be removed for weighing, sampling, and analysis.

VI. DURATION OF TESTS

38. The duration of tests shall be based on the firing of the minimum quantity of fuel to make the quantitative errors comparable for all tests. For both continuous and intermittent tests this quantity shall be considered as 10 times the hourly *continuous* rate of feed of the stoker at the setting of the feed being used, but in no case shall the total quantity of fuel fired be less than 200 lb per square foot of hearth or grate area.

VII. STARTING AND STOPPING OF CONTINUOUS TESTS

39. Before starting any test, if a different type or lot of fuel from that used in the preceding test is to be used, the stoker hopper shall be emptied and the feeding mechanism shall be flushed free of previous fuel. The feeding mechanism may be considered cleaned as the result of having it convey a volume of new fuel equal to twice the estimated gross volume of the feeding mechanism.

40. The fire shall be started by filling the feeding system with fuel fed from the hopper. A suitable amount of kindling shall be placed on the fuel in the furnace and ignited. The air supply shall be adjusted as recommended by the manufacturer or as indicated desirable by experience.

41. Operating Conditions. After the stoker has been started and the prescribed test conditions of rate of feed, draft in furnace, and air supply have been adjusted, all operating conditions shall have remained reasonably constant for one hour before the test period is started. The preliminary period shall be not less than 4 hours' duration.

42. (a) Start of Test. In stokers with automatic ash removal, the ashpit or ash removal mechanism shall be thoroughly cleaned to remove ash and refuse from previous tests and from the preliminary period. In stokers operating on the principle of removal of ash as clinker, any clinker that shall have been formed shall be removed. At the end of the preliminary period and after cleaning the fire the thickness and condition of the fuel bed shall be observed and recorded.

(b) The time, feed water, and boiler gage glass level shall be noted and recorded. The fuel in the stoker hopper shall be leveled off and the height recorded. The test shall start at the time of making these observations. A pail shall be placed under the separator outlet.

43. (a) End of Test. At the end of the test the fuel bed shall be as closely as possible of the same thickness and condition as at the start. After the clinker has been removed, if the fuel bed should be heavier than at the start, it is permissible to reduce the coal feed or to increase the air supply for a time necessary to burn the bed down to the thickness at the start. If it is thinner, it is permissible to reduce the rate of air supply for the time necessary to build the fuel bed up.

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(b) When the proper condition of the fuel bed is reached, the water level in the boiler and the level of the fuel in the stoker hopper shall be adjusted to the same height as at the start; the time shall be recorded and this time shall be the time of stopping. The feed water shall be quickly measured and recorded, and the separator water in the pail shall be weighed and recorded.

(c) In stokers operating with continuous or automatic ash removal, the ash shall be removed and weighed before preparation of the sample.

VIII. STARTING AND STOPPING OF INTERMITTENT TESTS

44. (a) Measurement and Procedure. In general, the same measurements and procedure must be followed in these tests as outlined in Section VII. Due to the fact that intermittent operation includes a heating and cooling of the boiler, it is absolutely necessary to be sure the same conditions exist at both the beginning and end of the test.

(b) These tests shall be started and stopped at the end of an *on* period.

(c) The intervals of stoker operation may be controlled by a suitable method such as a pressure-actuated switch operating between suitable limits, such as ¼ lb and 2 lb, by manually or automatically operated switch at definitely timed intervals, or by such other method as is indicated by conditions of the test.

IX. METHOD OF CONDUCTING TESTS

45. The steam pressure during continuous tests shall be maintained at a constant value. In intermittent tests, the steam pressure will vary. Enough readings of the pressure shall be taken at regular intervals in each *on* and *off* period to obtain a true average. A recording pressure gage is strongly recommended for intermittent tests. The water level in the boiler shall be maintained constant during the test.

46. Water Fed or Condensed. The records of water fed or condensed shall be made as required by the measuring system. It is desirable to measure the water fed or condensed each half-hour to check the rate of steaming.

47. Observations of drafts, pressures, and temperatures during *continuous tests* shall be made at regular intervals of not more than 15 min. During *intermittent tests* the flue-gas temperature shall be read a sufficient number of times, at regular intervals, in each *on* and *off* period to obtain a true average. A flue-gas temperature recorder is strongly recommended for intermittent tests. Separator water shall be weighed hourly.

48. Flue gas analyses shall preferably be made on samples collected for periods not exceeding 30 min. In intermittent tests, the gas shall be collected and analyzed separately for the *on* and *off* periods of operation unless a recorder is used. The samples may be held over several successive periods and analyzed hourly.

49. Smoke Readings. When the Ringelmann Chart Method is used, smoke observations shall be made every 30 sec through 1 hour in a continuous test or four complete *on* and *off* cycles in an intermittent test.

50. Attention to Fire. For stokers feeding not more than 60 lb of fuel per hour, intended for automatic operation, no attention shall be given to the fire during test periods other than the necessary removal of clinker or ash. For stokers feeding more than 60 lb of fuel per hour, attention may be given as required for satisfactory operation. The time and nature of each attention shall be recorded; the duration of each cleaning period shall be recorded.

X. PERFORMANCE CHARTS

51. (a) If four or more continuous tests are run at different combustion rates, a performance chart may be made. Against output expressed in thousands of Btu

and square feet of steam radiation, the following items may be plotted: stoker and boiler efficiency, flue-gas temperature, combustion rate, unit combustion rate, furnace draft, smokehood draft, windbox pressure, and CO_2. One square foot of steam radiation is defined as 240 Btu per hour.

(b) If four or more intermittent tests are run at varying intervals of operation, a performance chart may be drawn plotting stoker and boiler efficiency against output expressed in thousands of Btu and square feet of steam radiation.

STANDARD FORM

Table 1. Data and Results of Tests

General Information

1. Number of test
2. Date of test
3. Test conducted by
4. Type of test
5. Make and catalog designation of stoker
6. Type of stoker
7. Maximum feed rate of stoker
8. Make and catalog designation of boiler
9. Type of boiler
10. Commercial rating boiler.....sq ft steam radiation
11. Heating surface of boiler.....sq ft
12. Furnace volume.....cu ft
13. Hearth or grate dimensions, diam.....width.....length.....ft in.
14. Hearth or grate area.....sq ft
15. Weight of water in boiler to normal water line.....lb
16. Weight of water in boiler.....full.....lb
17. Water space.....per cent
18. Steam space.....per cent
19. Duration of test.....hours
20. Feed rate setting.....lb per hour
21. Barometer reading.....in. mercury

Fuel

22. Kind.....Seam.....Mine.....County.....State.....
23. Size

Proximate Analysis, as Fired

24. Moisture.....per cent
25. Volatile matter.....per cent
26. Fixed carbon.....per cent
27. Ash.....per cent
28. Calorific value, as fired.....Btu per lb

Ultimate Analysis, as Fired

29. Carbon.....per cent
30. Hydrogen.....per cent
31. Oxygen.....per cent
32. Nitrogen.....per cent
33. Sulphur.....per cent
34. Ash.....per cent

General Information

35. Total fuel fired during test period.....lb
36. Power consumed by stoker.....kwhr per ton of fuel fired
37. Average interval between attention to fire.....hour.....min
38. Average duration of cleaning period.....min

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Ash and Refuse

39. Total weight of ash and refuse ..... lb
40. Combustible in ash and refuse ..... per cent
41. Carbon burned per pound of fuel, as fired ..... lb

Drafts and Pressures

42. Stoker windbox pressure ..... in. water
43. Draft in furnace ..... in. water
44. Draft at boiler outlet ..... in. water

Flue Gases

45. Carbon dioxide (CO_2) ..... per cent
46. Oxygen (O_2) ..... per cent
47. Carbon monoxide (CO) ..... per cent
48. Nitrogen (N_2) (by difference) ..... per cent
49. Temperature of flue gases leaving boiler ..... F
50. Dry flue gas per pound of fuel, as fired ..... lb
51. Excess air ..... per cent
52. Temperature of air for combustion, dry-bulb ..... F

Steam and Feed Water

53. Steam pressure by gage ..... lb per sq in.
54. Absolute steam pressure ..... lb per sq in.
55. Temperature of feed water ..... F
56. Total weight of feed water ..... lb
57. Total weight of water from separator ..... lb
58. Moisture in steam ..... per cent
59. Total water evaporated ..... lb

Hourly Rates

60. Fuel fired (average) ..... lb per hour
61. Combustion rate (average) ..... lb per hour
62. Heat release rate (average) ..... Btu per hour
63. Unit combustion rate (average) ..... lb per sq ft of hearth or grate area per hour
64. Unit heat release rate (average) ..... Btu per cu ft furnace volume per hour
65. Actual water evaporated ..... lb per hour
66. Correction for uncovered boiler ..... lb per hour
67. Total corrected evaporation ..... lb per hour

Output

68. Output ..... Btu per hour
69. Output ..... sq ft steam radiation
70. Heat absorbed by boiler heating surface ..... Btu per sq ft per hour
71. Corrected evaporation per pound of fuel, as fired ..... lb

Heat Balance

	a BTU PER LB FUEL	b PER CENT
72. Heat transferred to boiler (efficiency of boiler and stoker)	.....	.....
73. Heat carried away by steam in flue gases	.....	.....
74. Heat carried away by dry flue gases	.....	.....
75. Heat lost by not burning carbon monoxide	.....	.....
76. Heat lost by not burning combustible in ash	.....	.....
77. Heat lost due to radiation and unaccounted for	.....	.....
Total	.....	100.0

CALCULATIONS FOR TABLE 1

Wherever CO_2, O_2, CO and N_2 are used they are percentages by volume of these constituents in the gases of combustion.

A_s = The allowed projected area of uncovered metal steam-heated surface.
H = Total heat (in Btu per lb) of dry saturated steam at the boiler outlet pressure.
h = Total heat (in Btu per lb) in feed water at boiler inlet.
t_s = Temperature of steam at boiler outlet pressure.

Formulae and Notes

$$\text{Item 17} = \frac{\text{Item 15}}{\text{Item 16}} \times 100$$

$$\text{Item 18} = \frac{\text{Item 16} - \text{Item 15}}{\text{Item 16}} \times 100$$

Item 36 = Total power consumed during test divided by total fuel fired expressed in tons.

Item 37 = Total time of test divided by number of attention periods.

Item 38 = Sum of duration of cleaning periods divided by the number of cleaning periods.

$$\text{Item 41} = \frac{\text{Item 29}}{100} - \frac{\text{Item 39} \times \text{Item 40}}{\text{Item 35} \times 100}$$

$$\text{Item 50} = \frac{700 + 4CO_2 + O_2}{3(CO_2 + CO)} \times \text{Item 41}$$

$$\text{Item 51} = \frac{(O_2 - \tfrac{1}{2}CO)}{0.264N_2 - (O_2 - \tfrac{1}{2}CO)} \times 100$$

Item 54 = (Barometric Pressure in Inches of Mercury $\times$ 0.491) + Item 53.

Item 56 = (When condenser is used) weight of water from condenser + Item 57.

$$\text{Item 58} = \frac{\text{Item 57}}{\text{Item 56}} \times 100$$

Item 59 = Item 56 − Item 57

$$\text{Item 60} = \frac{\text{Item 35}}{\text{Item 19}}$$

$$\text{Item 61} = \text{Item 60} - \frac{\text{Item 39} \times \text{Item 40} \times 14{,}600}{\text{Item 28} \times 100 \times \text{Item 19}}$$

Item 62 = Item 61 $\times$ Item 28

$$\text{Item 63} = \frac{\text{Item 61}}{\text{Item 14}}$$

$$\text{Item 64} = \frac{\text{Item 62}}{\text{Item 12}}$$

$$\text{Item 65} = \frac{\text{Item 59}}{\text{Item 19}}$$

$$\text{Item 66} = \frac{1.8}{H\text{-}h}(t_s - \text{Item 52})A_s$$

Item 67 = Item 65 + Item 66

Item 68 = Item 67 $(H\text{-}h)$

$$\text{Item 69} = \frac{\text{Item 68}}{240}$$

$$\text{Item 70} = \frac{\text{Item 68}}{11}$$

$$\text{Item 71} = \frac{\text{Item 67}}{\text{Item 60}}$$

$$\text{Item } 72a = \frac{\text{Item } 68}{\text{Item } 60}$$

$$\text{Item } 73a = 9 \times \frac{\text{Item } 30}{100} \times (1090.7 + 0.455 \times \text{Item } 49 - \text{Item } 52)$$

$$\text{Item } 74a = \text{Item } 50 \times 0.24\ (\text{Item } 49 - \text{Item } 52)$$

$$\text{Item } 75a = \frac{CO}{CO_2 + CO} \times \text{Item } 41 \times 10{,}150$$

$$\text{Item } 76a = \left(\frac{\text{Item } 29}{100} - \text{Item } 41\right) \times 14{,}600$$

$$\text{Item } 77a = \text{Item } 28 - (\text{Item } 72 + \text{Item } 73 + \text{Item } 74 + \text{Item } 75 + \text{Item } 76)$$

$$\text{Items } 72b \text{ to } 77b, \text{ inclusive} = \frac{\text{Items } 72a \text{ to } 77a, \text{ inclusive}}{\text{Item } 28} \times 100$$

SHORT TEST FORM

Table 2. Data and Results of Tests

General Information

1. Number of test
2. Date of test
3. Test conducted by
4. Type of test ..... continuous ..... intermittent
5. Make and catalog designation of stoker
6. Type of stoker
7. Maximum feed rate of stoker
8. Make and catalog designation of boiler
9. Type of boiler
10. Commercial rating of boiler ..... sq ft steam radiation
11. Heating surface of boiler ..... sq ft
12. Furnace volume ..... cu ft
13. Hearth or grate dimensions, diam ..... width ..... length ..... ft in.
14. Hearth or grate area ..... sq ft
15. Weight of water in boiler to normal water line ..... lb
16. Weight of water in boiler ..... full ..... lb
17. Water space ..... per cent
18. Steam space ..... per cent
19. Duration of test ..... hours
20. Feed rate setting ..... lb per hour
21. Barometer reading ..... in. mercury

Fuel

22. Kind ..... Seam ..... Mine ..... County ..... State
23. Size
24. Calorific value, as fired ..... Btu per lb
25. Total fuel fired during test period ..... lb
26. Average interval between attention to fire ..... hour min
27. Average duration of cleaning period ..... min

Ash and Refuse

28. Total weight of ash and refuse ..... lb
29. Combustible in ash and refuse ..... per cent

Drafts and Pressures

	On Periods	Off Periods	
30. Stoker windbox pressure			in. water
31. Draft in furnace			in. water
32. Draft at boiler outlet			in. water

Flue Gases

	On Periods	Off Periods	
33. Carbon dioxide (CO_2)			per cent
34. Temperature of flue gases leaving boiler			F
35. Temperature of air for combustion, dry-bulb			F

Steam and Feed Water

36. Steam pressure by gage	lb per sq in.
37. Absolute steam pressure	lb per sq in.
38. Temperature of feed water	F
39. Total weight of feed water	lb
40. Total weight of water from separator	lb
41. Moisture in steam	per cent
42. Total water evaporated	lb

Hourly Rates

43. Fuel fired (average)	lb per hour
44. Combustion rate (average)	lb per hour
45. Heat release rate (average)	Btu per hour
46. Unit combustion rate (average)	lb per sq ft of hearth or grate area per hour
47. Unit heat release rate (average)	Btu per cu ft furnace volume per hour
48. Actual water evaporated	lb per hour
49. Correction for uncovered boiler	lb per hour
50. Total corrected evaporation	lb per hour

Output

51. Output	Btu per hour
52. Output	sq ft steam radiation
53. Heat absorbed by boiler heating surface	Btu per sq ft per hour
54. Corrected evaporation per pound of fuel as fired	lb

Efficiency

55. Heat transferred to boiler (efficiency of boiler and stoker)	per cent

CALCULATIONS FOR TABLE 2

Wherever CO_2, O_2, CO and N_2 are used they are percentages by volume of these constituents in the gases of combustion.

A_s = The allowed projected area of uncovered metal steam-heated surface.
H = Total heat (in Btu per lb) of dry saturated steam at the boiler outlet pressure.
h = Total heat (in Btu per lb) in feed water at boiler inlet.
t_s = Temperature of steam at boiler outlet pressure.

Formulae and Notes

$$\text{Item 17} = \frac{\text{Item 15}}{\text{Item 16}} \times 100$$

$$\text{Item 18} = \frac{\text{Item 16} - \text{Item 15}}{\text{Item 16}} \times 100$$

Item 26 = Total time of test divided by number of attention periods.

Item 27 = Sum or duration of cleaning periods divided by the number of cleaning periods.

Item 37 = (Barometric Pressure in Inches of Mercury $\times$ 0.491) + Item 36.

Item 39 = (When condenser is used) weight of water from condenser + Item 40.

$$\text{Item 41} = \frac{\text{Item 40}}{\text{Item 39}} \times 100$$

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$$\text{Item } 42 = \text{Item } 39 - \text{Item } 40$$

$$\text{Item } 43 = \frac{\text{Item } 25}{\text{Item } 19}$$

$$\text{Item } 44 = \text{Item } 43 - \frac{\text{Item } 28 \times \text{Item } 29 \times 14{,}600}{\text{Item } 24 \times 100 \times \text{Item } 19}$$

$$\text{Item } 45 = \text{Item } 44 \times \text{Item } 24$$

$$\text{Item } 46 = \frac{\text{Item } 44}{\text{Item } 14}$$

$$\text{Item } 47 = \frac{\text{Item } 45}{\text{Item } 12}$$

$$\text{Item } 48 = \frac{\text{Item } 42}{\text{Item } 19}$$

$$\text{Item } 49 = \frac{1.8}{H\text{-}h}\,(t_s - \text{Item } 35)\,A_s$$

$$\text{Item } 50 = \text{Item } 48 + \text{Item } 49$$

$$\text{Item } 51 = \text{Item } 50\,(H\text{-}h)$$

$$\text{Item } 52 = \frac{\text{Item } 51}{240}$$

$$\text{Item } 53 = \frac{\text{Item } 51}{\text{Item } 11}$$

$$\text{Item } 54 = \frac{\text{Item } 50}{\text{Item } 43}$$

$$\text{Item } 55 = \frac{\text{Item } 51 \times 100}{\text{Item } 43 \times \text{Item } 24}$$

No. 1091

A TEST METHOD FOR AIR FILTERS *

By Richard S. Dill ** (*NON-MEMBER*), Washington, D. C.

THE apparatus now used for testing air filters at the National Bureau of Standards consists of a means of drawing air through a specimen filter at a predetermined rate, a means of impregnating the air with dust before it reaches the filter under test and a means of drawing measured samples of the filtered and unfiltered air simultaneously through filter papers. A photometer which employs a source of light and a photo-electric cell is provided for comparing the densities of the dust deposits on the filter papers. At present the standard test dust is a fly ash called Cottrell precipitate since it is obtained from a Cottrell precipitator in a local power plant burning pulverized coal. Whatmans No. 41 filter paper has been found satisfactory and has been used during the tests.

The general arrangement of the apparatus is shown in Fig. 1. The air ducts shown and the elutriating tank are made of 24 gage galvanized iron. Air enters through the duct connecting to the window, as indicated at the top right of Fig. 1, and is discharged from the lower pipe through the same window.

The entering air can be passed directly to the filter under test, as was formerly done for trials with atmospheric dust, or it can be shunted downward through the 10-in. pipe shown and forced to come upward through the elutriating tank before reaching the specimen.

The purpose of the elutriating tank is to arrest abnormally large particles and prevent them from reaching the filter.

There are flow straighteners consisting of nests of 4-in. gutter pipe in the elutriating tank and in the horizontal duct between it and the filter.

Leaving the elutriating tank the air passes through the specimen filter and the metering orifice, goes down through a vertical pipe, passes through the blower and is ejected out of doors through the lower duct.

A telescoping section in the main duct permits filters up to about 30 in. in length to be accommodated. Since filters differ in size and shape a wooden or sheet metal duct section is usually constructed to receive the filter and the duct section is fitted in the space provided in the main duct.

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* Publication approved by the Director of the National Bureau of Standards of the U. S. Department of Commerce.

** Associate Mechanical Engineer, National Bureau of Standards.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Hot Springs, Va., June, 1938.

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A slant gage is used to measure the pressure loss or resistance of the filter. A U-tube water manometer is used to indicate the pressure drop across the metering orifice and hence the air flow in the main duct. A 6-in. orifice is used for deliveries up to about 1200 cfm and a 9-in. orifice for flows up to 2000 cfm which is near the upper limit of capacity of the apparatus.

For taking samples of air to be passed through filter papers, sampling tubes are used, one on the clean air and one on the dirty air side of the filter under test. Their positions are indicated on Fig. 1 and the construction of the sample collector is shown on Fig. 2.

Each collector is made of 1½-in. brass tubing and is about 12-in. long. It contains a metering orifice about 0.18-in. in diameter. It is reduced in size and fitted with a half-inch standard pipe connection at the entrance end. The discharge end is covered by a cap held against the flat end of the tube

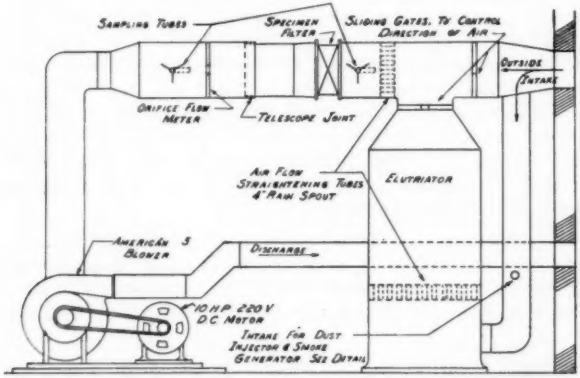

FIG. 1. DIAGRAM OF TESTING EQUIPMENT FOR AIR FILTERS

by a pair of springs. The springs can be stretched by hand to permit a filter paper clasp to be inserted between the cap and tube. The cap is connected by a rubber tube to a vacuum line. The filter paper clasp consists of two matching brass plates with an aperture across which the filter paper is stretched.

Each sampling tube consists of a 2-ft length of half-inch pipe which enters the main duct at right angles and turns on a 6-in. radius to face up-stream. Air is drawn from the main duct through the pipe and into the collector. In the collector the air passes first through the metering orifice, then through the filter paper and is discharged to the vacuum line.

A one-leg water manometer is used to indicate the air flow rate through the metering orifice in each collector. The manometers for the two collectors are mounted adjacent to each other for convenience and their scales are graduated to read air flow in cubic feet per minute. Their readings are called sampling rates and the sampling rates can be adjusted by needle valves in the vacuum lines.

The photometer used for comparing the density of dust deposits on filter papers consists essentially of an automobile headlight bulb and a photo-cell

with an aperture across which the filter paper can be placed so that the light from the bulb passes through the paper and falls on the cell. The aperture is, of course, smaller than any dust spot taken. The arrangement is shown on Fig. 3.

Originally the cell was connected directly to a galvanometer and the reading of the galvanometer was considered indicative of the intensity of the light

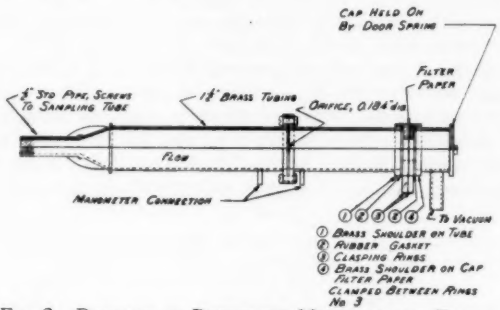

FIG. 2. DIAGRAM OF DEVICE FOR METERING AND FILTERING AIR SAMPLES

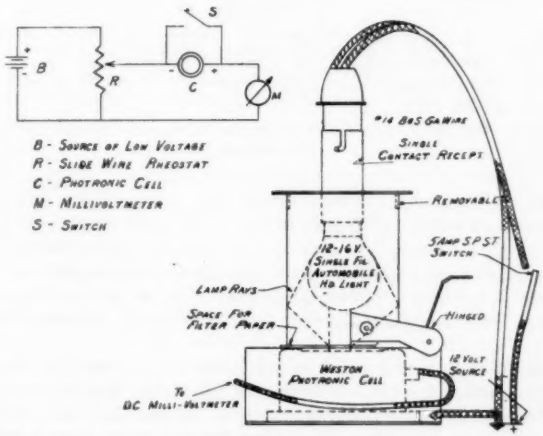

FIG. 3. DIAGRAM OF PHOTOMETER FOR MEASURING LIGHT TRANSMISSION OF FILTER PAPERS

striking the cell. This arrangement was improved by employing a circuit in which the effect on the cell of external resistance is made unimportant. The circuit is described in a paper by Wood.[1] The wiring diagram is shown on Fig. 3. In use, the source of low voltage acts in the same direction in producing current in the circuit as the photo-cell. The rheostat, R_1, is ad-

[1] Zero-Potential Circuit for Blocking-Layer Photo-Cells, by Lawrence A. Wood (*The Review of Scientific Instruments*, Vol. 7, No. 3, p. 157, March, 1936).

justed until opening and closing the switch makes no difference in the reading of the millivoltmeter. For this condition the cell is acting against no resistance external to itself so that the effect of changes in its internal conductivity is avoided. The readings of the millivoltmeter at this condition are accepted as an indication of the light passing through a filter paper and falling on the cell and, consequently, of the transparency of the paper.

The device for impregnating air with dust is shown in Fig. 4. It is located on the 10-in. pipe ahead of the elutriating tank in the system. The elements of the device are a drum, rotated slowly by a motor, a dust hopper above the drum arranged to lay a ribbon of dust on it, and an aspirator, operated by compressed air, arranged to draw the dust from the drum and inject it into the 10-in. pipe in which the main air stream is passing on its way to the elutriating tank and filter. The dust injector is regulated to pass about one gram of dust per thousand cubic feet of air. Ten grams are

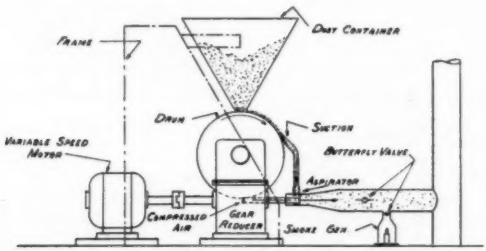

FIG. 4. DIAGRAM OF DUST INJECTOR AND SMOKE GENERATOR

charged into the hopper for a run so that the duration of a run for a filter with a delivery of 800 or 1000 cfm is 10 or 12 min.

The use of this quantity of dust at this rate has been found to result in dust spots on the filter papers of good density for convenient comparison.

Air is drawn through a filter mounted for test at the rate specified for it by its maker. Its resistance is noted and trials of its efficiency are made at this rate.

Present practice during trials for efficiency is to draw samples of air at equal rates, usually 1 cfm, through the upstream and downstream filter papers. The spot on the downstream paper exposed for the passage of air and hence for dust deposition is smaller than that on the upstream paper. If the area of the downstream spot is half that of the upstream spot and the spots are found to be equally dense, this means that the upstream air carried twice as much dust as the downstream air and the efficiency by this test method is therefore considered to be 50 per cent. If the downstream spot is found to be the cleaner, the efficiency is considered to be above 50 per cent, and vice versa.

The size of a filter paper spot is governed by the size of the aperture in the filter paper clasp. Pairs of clasps are provided having apertures corresponding to efficiencies of 50, 60, 65, 70, 75 and 80 per cent. The present

routine test consists of finding by trial that the efficiency of a specimen filter is above one of these values and below the next higher.

Evaluation of an efficiency with a precision of one or two per cent is possible. Such evaluations are not customarily made because integral efficiencies, multiples of five, have been chosen by the purchasing agencies of the Government as criteria for the performance of the various classes of filters. Filters with efficiencies below 60 per cent are not acceptable and filters with efficiencies above 80 per cent are generally acceptable under present Government specifications.

In preparation for a run all parts of the apparatus are swept out by drawing air through them at a comparatively high rate. A pair of filter papers, previously found to be photometrically similar, is placed in a pair of clasps and these are placed in the spaces provided in the sample collectors. Ten grams of dust are put in the dust injector. Flow rates through the filter and through the sampling tubes are adjusted and the dust injector is then started. The run continues until all the dust has been injected, at which time the flow through the sampling tubes is cut off and the papers are removed for photometric examination. Successive runs are made until the efficiency is evaluated as indicated above.

The apparatus was installed at the Bureau in 1934 at the request of the Procurement Division of the Treasury Department and it has been used since that time without essential change as a type test for air cleaning devices offered for sale to the Government.

At the outset, tests of filters with atmospheric dust, that is dust normally in the outside air on a clear day at the Bureau of Standards, were attempted. It was found, however, that the efficiencies of filters were uniformly low and difficult to evaluate with this dust. Trials with smoke from a fuel oil flame showed that this substance was subject to the same objections, and therefore tests with these two dusts are no longer commonly made.

With the Cottrell precipitate, however, efficiencies of filters on the market were found to be spread over a wide range, specifically from less than 10 to something over 80 per cent and the differences between filters are more easily shown.

A variation of the test method has been used as a basis for a specification for electro-static dust precipitating equipment. The specification required that the equipment have an efficiency of 90 per cent or more on atmospheric dust on any clear day. The specified test consisted of drawing air at equal sampling rates through filter paper spots of equal area on the two sides of the equipment, and drawing air through the downstream spot ten times as long as through the upstream spot. The requirement was that under this condition the downstream spot should not be the denser.

Means of improving the test are being sought. Some efforts to separate the dust into component size ranges by elutriation have shown promise but have not been carried to a conclusion.

An efficiency test apparatus of different design has been tried, also with promising results. This apparatus consists essentially of a means of drawing samples of air through two filter papers simultaneously while light from a single bulb passes through the two papers and falls on two photo-cells.

The cells are connected to the same galvanometer and if the light falling on one cell decreases faster than the light falling on the other, the galvanometer

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moves off the zero position. In making a test, the flows of air through the filter papers are kept in such ratio as to keep the galvanometer on the zero position, showing that the papers are becoming dirty at equal rates. The ratio of the sampling rates is then an index to the efficiency of the filter under test. With this apparatus a continuous record of the efficiency of a filter over a period of time is possible.

The test method as it is now applied gives information on the performance of filters when new. It gives no information on the effects of time or use on performance but its application has permitted the selection of filters with useful initial effectiveness and this, in considerable degree, puts the selection of filters on a definite basis. The resistance of filters commonly increases with dust load and it is probably in order for the test to be extended or altered to obtain information on this point. However, such information is held to be of secondary importance compared to knowledge of the ability of filters to arrest dust, since arresting dust is the prime purpose of a filter.

DISCUSSION

ARTHUR NUTTING[2] (WRITTEN): Mr. Dill of the Bureau of Standards deserves much credit for the great amount of work he and the Bureau have done in the developing of the photometric method of testing air filters. They have not only made a greater progress than any other operators in the country in the development of this method, but they have put the method to practical use and have used it constantly. I believe, however, that there are several criticisms as they employ it which are in order, but nevertheless the method, even as it exists now, can be manipulated to place a value on air filter performance, which value should be comparable to that obtained by the A.S.H.V.E. method. The chief advantages of this method are that readings can be taken faster, with more accuracy.

This method, as the Bureau of Standards employs it, however, is concerned only with the initial cleaning efficiency of the filter. The manufacturers of air filters design their filters not with the idea of obtaining a high cleaning efficiency on the first few hours' operation, but with the idea of obtaining the highest average cleaning efficiency. The Bureau of Standards' tests are always made by taking three or four cleaning efficiency readings, each reading requiring 10 or 12 min of dust feeding. A total of 40-50 g (grams) of dust is usually fed during these tests—a viscous filter will have a dust holding capacity of approximately 500-700 g, on Cottrell dust. The cleaning efficiencies prescribed to the filter by the Bureau, therefore, are invariably taken at the lowest point on the cleaning efficiency curve of the filter and are no indication of the maximum or the average cleaning efficiency of the filter. Further, it is considered good economy to design a filter to hold a large quantity of dust, even though a few per cent in cleaning efficiency must be sacrificed. This feature of the filter, as Mr. Dill mentions, is entirely neglected by the Bureau of Standards' method, and the result is that a manufacturer could pass the Bureau of Standards' specifications by furnishing a filter which had a high initial cleaning efficiency but which would remain in service only a very short period of time. If, in making these tests, a simple change of loading the filter with, say, 100 g of dust before making any of the cleaning efficiency tests was made, an answer more truly representative of the value of the filter would be obtained.

There has been a great tendency in the past two or three years to test air filters by some kind of discoloration method. In the field in some cases it has been the practice to place cheese cloth targets behind filters and examine the discoloration

[2] Chief Engr., American Air Filter Co., Inc., Louisville, Ky.

visually. To date, there is no information which would allow one to express a discoloration method in terms of a weight method of testing air filters, and I believe it would be very interesting for the Society to conduct such research with some kind of photo-metric test operating simultaneously with the standard A.S.H.V.E. method.

G. W. PENNEY (WRITTEN): There are many uses for air cleaning devices, so that a device which is satisfactory in one application may be of little value in another. The making of adequate tests is difficult. If we could test a device with a dust made up of a wide range of materials and particle sizes, and could determine the efficiency on each of the many kinds and sizes of particles, the data would be quite satisfactory. However, such a dust is almost never practical and simpler tests must be devised.

In some industrial applications where we are interested in recovering valuable materials, a weight test is probably the best measure of efficiency. However, in air

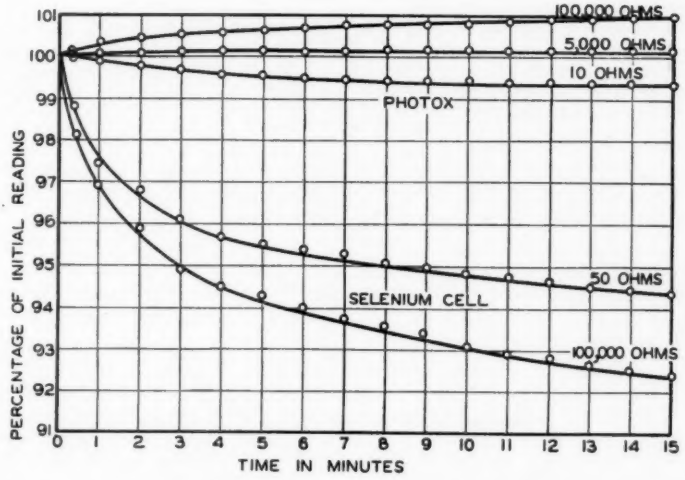

FIG. A. INITIAL DRIFT OF PHOTOX AT 50 FOOT CANDLES AS COMPARED TO THAT OF A SELENIUM CELL

conditioning we usually are interested in keeping a building clean and in this case, a weight test has little meaning. For example, in one method, the test dust is sifted through a 200-mesh screen which permits particles up to 0.0025 in. or 63 μ (micron) in diameter. However, measurements of actual atmospheric pollution have shown that most of the particles are well under ½ μ in diameter. One 63 μ particle represents as much volume as 2,000,000 one-half μ particles. If we then imagine a dust made up of 2000 one-half μ particles for each 63 μ particle, removing the large particle accounts for 99.9 per cent of the weight, but 1/20 of one per cent of the number of particles.

There are several variations of this test which will give slightly different results. For example, instead of using the same elapsed time and different velocities for the filtered and unfiltered air, we might use the same velocities through each sample but increase the time on the filtered samples. This seems to be a little more accurate measure of efficiency, but is difficult to make because the dust must be fed at a per-

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fectly constant rate. Since Mr. Dill's method is the more simple and reliable method, I am very much in favor of the method as presented.

I would like to emphasize the importance of two paragraphs toward the end of the paper where Mr. Dill points out that the type of dust may be varied so as to give a wide range in efficiency, and that some devices, which gave a reasonably high efficiency on the Cottrell precipitate, when tested on the atmospheric dust normally in the air, gave efficiencies so low that it was difficult to assign a numerical efficiency.

In the photometer shown in Fig. 3, a photronic cell is used. This photocell is recognized as having the most permanent calibration (i.e. low aging), but this is of no consequence in this application, since the light is adjusted each time to give a standard reading on the meter. In this application this cell has a serious disadvantage, because it gives considerable *initial drift* or *fatigue*, necessitating the special circuit shown in Fig. 3, and considerable time in reading. For this application, copper oxide cells are available which, while they age appreciably, have a negligible initial drift or fatigue shown in Fig. A, and can, therefore, be connected directly to a micro-ammeter and read almost instantly.

In conclusion, I think Mr. Dill is to be congratulated for presenting a very simple method of testing, which measures the quality of blackness. For air conditioning applications, this blackness is usually of first importance. In making the test, care should be taken to insure that the test dust used is representative of the type of dust to be removed in service.

No. 1092

THE FLOW OF AIR THROUGH EXHAUST GRILLES

By A. M. GREENE, JR.* (*NON-MEMBER*) AND M. H. DEAN ** (*MEMBER*)
PRINCETON, N. J.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Princeton University.

THE primary object of this research problem was the determination of a coefficient or series of coefficients to be used in calculating the air flow through an exhaust grille of modern form from anemometer readings.

It will be recalled that this subject was treated by Davies in papers [1] previously presented before the Society in which he proposed the following formula for exhaust grilles:

$$Cfm = K\,V\,A \qquad (1)$$

in which:

V = Corrected velocity from anemometer in feet per minute.
A = Gross area in square feet (duct size).
K = An average coefficient dependent on velocity.

For K, Professor Davies found average values ranging from 0.762 at low velocities to 0.832 at high velocities, the mean of these being 0.80.

With the increase in the per cent free area of grilles in recent years, the applicability of this factor for grilles of more modern design has been questioned. This paper reports tests to determine the applicability of the earlier formula to recent designs of grilles.

The results of this research indicate that a simple general formula for air discharge to be used with exhaust grilles of recent design within a maximum error of ± 6 per cent is

$$Cfm = 0.873\,V\,A \qquad (2)$$

where:

* Dean and Professor of Mechanical Engineering, School of Engineering, Princeton University.
** Dean-Hagny Corp., Kansas City, Mo., formerly Graduate Student, Princeton University.

[1] The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies. (A.S.H.V.E. TRANSACTIONS: Vol. 36, 1930, p. 201; Vol. 37, 1931, p. 619; Vol. 39, 1933, p. 373.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by M. H. Dean.

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V = Velocity in feet per minute indicated by corrected anemometer readings.
= Anemometer reading for complete moving traverse with automatic weighted value divided by total time and corrected by anemometer calibration.
A = Gross frontal area = the smallest area of the inside of the frame of the grille in square feet.

The effects of size of grille and velocity with a smaller possible maximum error are eliminated by the use of a coefficient K in place of the average coefficient 0.873. K is determined by the following formula:

$$K = \frac{0.904}{R^{0.04}} \left(1 + \frac{V - 600}{40{,}000}\right) \quad \ldots\ldots (3)$$

where:

K = Coefficient in the formula $\text{Cfm} = K\,V\,A$

R = Hydraulic radius of duct in inches = $\dfrac{\text{Normal Area in square inches}}{\text{Nominal Perimeter in inches}}$

V = Velocity in feet per minute as indicated by anemometer.

The word *nominal* refers to the dimensions of the duct into which the grille under consideration fits.

The results of this research indicate the necessity of making measurements in a definite manner and with definite standard equipment and also the desirability of formulating a code for the determination of the air discharge through exhaust grilles.

DESIGN OF TEST EQUIPMENT

The following five points were considered in the design of the testing equipment for exhaust grilles:

1. A simple method must be employed to measure the true quantity of air passing through the system.
2. There must be an easy method of changing the size and type of grille.
3. An air-tight system is imperative.
4. The air volume must be controlled.
5. The air must not be in an eddy condition upon entering the measuring device.

Keeping these points in mind the equipment was designed as shown in Figs. 1 and 2.

The duct of 12 in. inside diameter was connected to a 4 ft x 4 ft x 6 ft plenum chamber. This chamber acted as a stilling box for the air since the cross-sectional area was very large in comparison to the free area of the grille. It also served as an excellent device for mounting the grilles, which were set in pieces of plywood of a size to fill the opening in the end of the plenum chamber. The plywood was mounted on the plenum chamber by means of bolts through the sheet metal and backed by a metal frame of angle irons. Soft putty was filled in around the board to prevent any possible air leakage. All other joints in the system were soldered.

The fan was driven by a variable speed motor and the fan discharge area was variable.

It will be noted in Fig. 2 that the air left the plenum chamber through a large sheet metal nozzle of throat diameter equal to the duct diameter.

This prevents irregular flow due to the contraction of air passage from the area of the plenum chamber to that of the duct. Straightening vanes of the *egg crate* type were made of 2-in. squares 9 in. deep.

The nozzle for the measurement of the flow of air, shown in Figs. 3 and 4, was of the elliptical profile type. It was made of dimensions identical to those of the *U. S. Bureau of Standards* Nozzle C-2.[2]

THE NOZZLE AND ITS CALIBRATION

The authors were somewhat at a loss to choose the best method of measuring the quantity of air. A water displacement tank was suggested and discarded because of cost and difficulty in handling. The choice then remained between a venturi meter, a nozzle, a flat plate orifice, and a Pitot

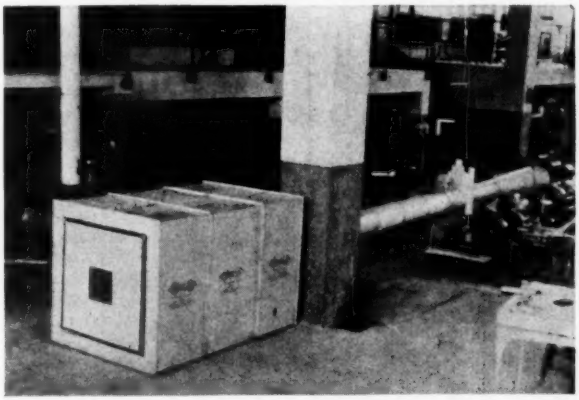

FIG. 1. ARRANGEMENT OF APPARATUS

tube. Of these types the nozzle was chosen because of its accuracy and the simplicity of mounting it in the 12-in. duct. Although many recent papers have dwelt upon the value of the flat plate orifice, its coefficients have greater limits of variation than those of the nozzle or the venturi meter. The Pitot tube, successfully employed by Davies, would have been an ideal method of measuring the air but its use was dispensed with since 20 grilles were to be tested at various air velocities and duct Pitot tube traverses and anemometer grille traverses could not be taken at the same time by a single observer. Therefore, it was decided to use a metering device which gave the quantity of air flow from a single reading.

In order to have a check on the nozzle coefficient, a nozzle which had been built and calibrated was followed in design. As stated before, this nozzle was the type C-2 of the *U. S. Bureau of Standards* Nozzle for which the coefficient varies from 0.993 to 0.995.[3] The results of the calibration of

[2] *U. S. Bureau of Standards, Journal of Research*, Vol. 2, 1929, No. 3, p. 561.
[3] Loc. Cit. See Note 2, p. 652.

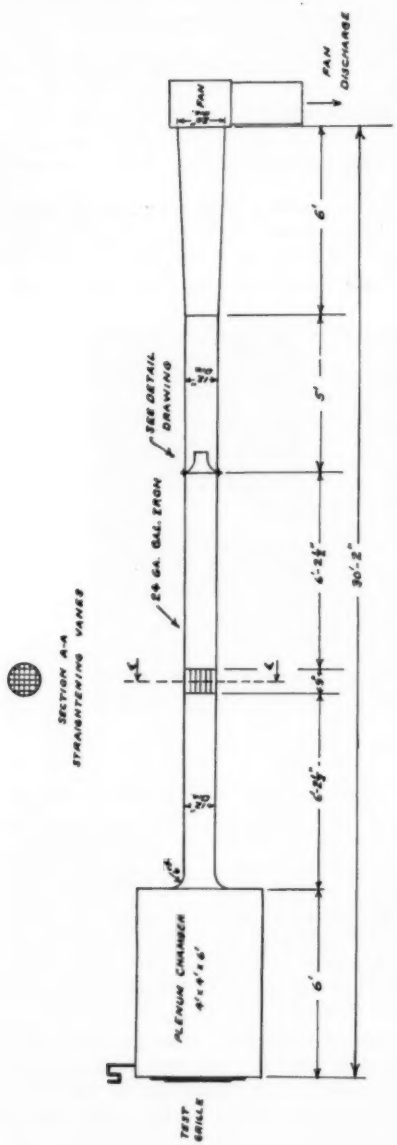

FIG. 2. ARRANGEMENT OF APPARATUS

the nozzle used in the tests described in this paper are seen in Fig. 5 to vary from 0.994 to 0.999, which gives a maximum difference of four-tenths of one per cent from the results obtained by the Bureau of Standards. The pressure drop was measured through 1/16 in. flush holes at the plane of the outlet to the nozzle and at 17½ in. upstream from this point.

The calibration was performed by means of a Standard A.S.H.V.E. Pitot tube using a coefficient of unity as should be employed where the air at the nozzle throat or point of Pitot tube traverse is in streamline flow.

Four Pitot tube readings were taken at quartering points in the mid-area points of three imaginary concentric sections of equal areas formed in the 5-in. throat diameter. The locations of these readings are shown in Fig. 6.

In using the Pitot tube, the velocity determined from the gage reading at any point, x, applies to an element of cross-sectional area of $2\pi r_x dr$ so that the volume flowing per second through a circular area is given by

$$\text{Volume per Second} = 2\pi \int r_x V_x dr = \pi \int V_x d(r_x)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

It is seen from this equation that if the velocity is plotted as ordinate and

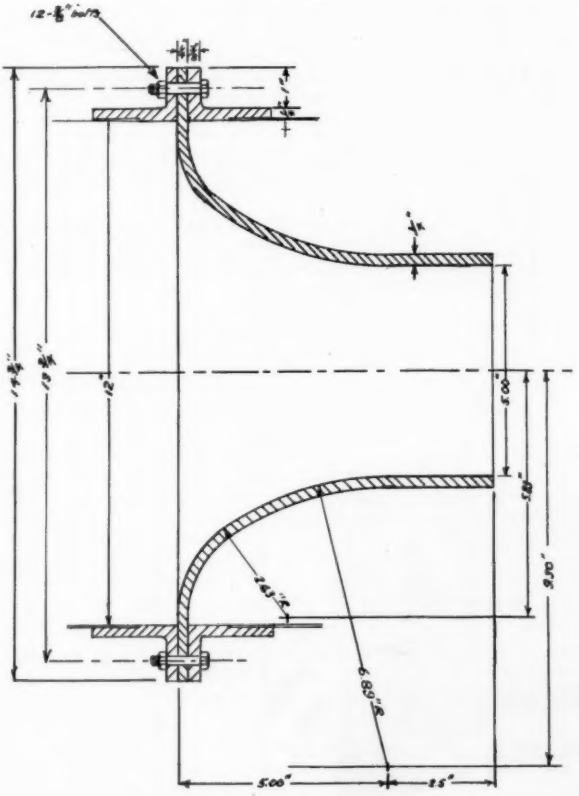

FIG. 3. SECTION THROUGH NOZZLE

the square of the radius as abscissa, the area under the curve when multiplied by π will give the discharge per second. Since the partial areas to which each set of Pitot tube readings refer are equal, this plotting is unnecessary and the average velocity for each set may be averaged to obtain the mean velocity through the nozzle. The velocity in feet per minute from the Pitot

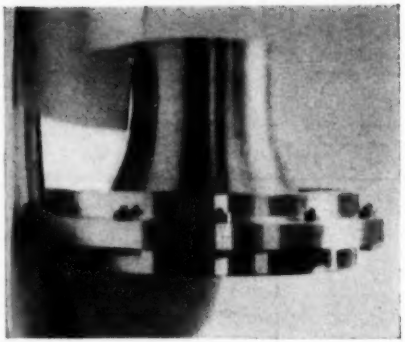

FIG. 4. NOZZLE

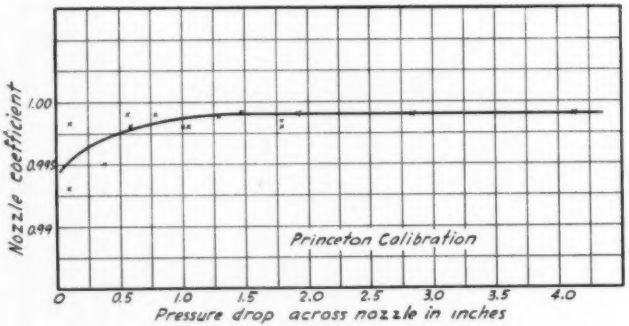

FIG. 5. COEFFICIENTS FOR NOZZLE

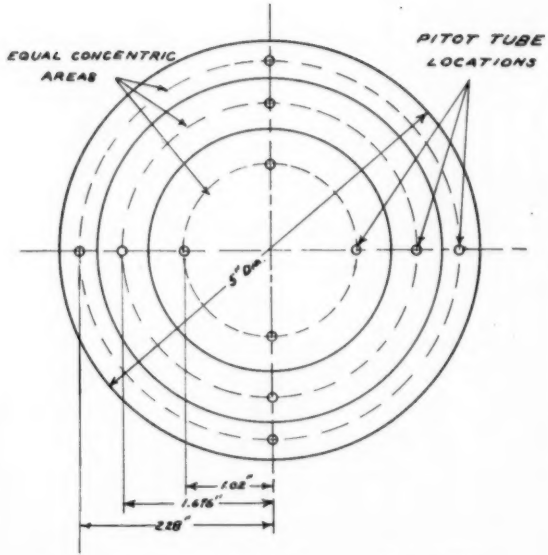

FIG. 6. LOCATIONS OF PITOT TUBE AT NOZZLE THROAT

tube reading, h, in inches of water, and with air of relative density, x, is given by Equation 5:[4]

$$V = 4006\sqrt{h/x} \quad \ldots\ldots(5)$$

The four readings of h of the Pitot tube in any zone were averaged, and the mean value of $\sqrt{h}$ for the full nozzle was found by extracting the square root of the average reading for each zone and then determining the average of the square roots of the average of the zone readings.

Since the Pitot tube readings apply to equal annular areas, the average $\sqrt{h}$ can be applied to the full area, and the true quantity of air delivered from the nozzle can be found by multiplying the average velocity by the outlet area of the nozzle.

The quantity of air at outlet flowing through the nozzle as indicated by the pressure drop is given by formula 6:[5]

$$\text{Volume per Second} = 8.02A_2\sqrt{\frac{\triangle p \times v_1}{1 - \left(\frac{A_2}{A_1}\right)^2\left(1 - 1.428\,\frac{\triangle p}{p_1}\right)}} \quad \ldots\ldots(6)$$

in which:

A_2 = Nozzle throat area in square feet.
A_1 = Duct area in square feet.
$\triangle p$ = Pressure drop from point p_1 above nozzle to point p_2 following nozzle in pounds per square foot.
v_1 = Volume of 1 lb of air under conditions of test at entrance.

Having solved for the quantity as determined by the Pitot tube and for that indicated by the pressure drop across the nozzle, the nozzle coefficient was determined at various velocities by dividing the Pitot tube quantity by the nozzle quantity. The result of this calibration is shown in Fig. 5.

The results of Pitot tube traverses demonstrate that the Pitot tube does have a noticeable interference with air flow. Although the velocity pressure is for the most part correct at a maximum in the center of the nozzle, the pressure drop across the nozzle, which should remain constant, has a gradual decrease as the Pitot tube is passed in front of the outlet. This is explained by the fact that the tube, upon insertion into the duct, sets up eddies and friction losses, and, therefore, the fan draws less air through the nozzle. It will be apparent that these values are not large enough to affect appreciably the accuracy of the work, yet they do exist and are here recorded with the idea of stimulating some additional work on the properties of Pitot tubes and the feasibility of using streamlined tubes.

In order to eliminate errors due to different gage characteristics, a single gage was used on any two sets of comparative readings. All readings below an inch of water were taken on a Wahlen gage (Fig. 7); whereas larger readings were taken on an inclined draft gage. These gages were also checked against one another and found to give identical results.

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[4] See Appendix No. 1.
[5] See Appendix No. 2.

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The density of the alcohol in the Wahlen gage was 0.8142, while the kerosene and ligroin mixture in the inverted U-tube had a density of 0.8048. The difference between these two densities is 0.0094, which is ideal for the operation of the gage. These densities were determined by means of the displacement method using gravimetric scales. The readings on the Wahlen gage were changed from inches of alcohol to inches of water by multiplying the gage reading by the density of the alcohol (0.8142).

METHOD OF ANEMOMETER TRAVERSE

Before making an attempt to determine the discharge coefficients for grilles, it was necessary to experiment with the various methods of taking a traverse

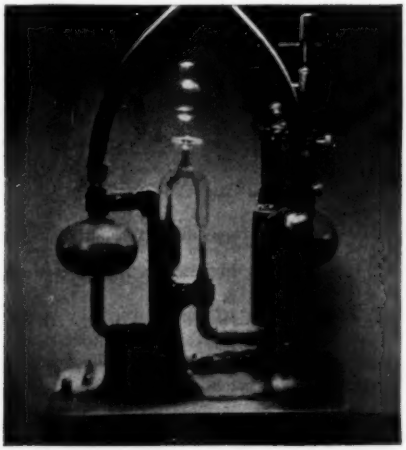

FIG. 7. WAHLEN GAGE

with an anemometer in contact with the grille. It was immediately discovered that the readings near the center of all types of grilles were much higher than those near the edges and it occurred to the observer that perhaps some allowance should be made for this variation.

A series of observations also showed that there was an appreciable effect due to the obstruction of the hand in holding the anemometer. To overcome this, a ⅜-in. hollow rod 2 ft long was attached to the top of the anemometer. Through the hole in the rod a stiff wire was connected to the starting and stopping catch on the anemometer, so that it was controlled at the end of the rod. Time was taken by an electric timer which rang a bell every 15 seconds, and, therefore, complete attention could be given to the making of the traverse. All tests were made using the electric timer and the rod-mounted anemometers, although several check runs were made using a stop watch. In making a continuous type traverse, it was found that 15 seconds was a convenient time for

making one crossing of both the 1- and 2-ft grilles. No advantage was gained by spending 30 seconds on the larger size.

Both a 3-in. and a 4-in. anemometer were used in all but the first few tests, and it was found that the instruments checked within two or three per cent of each other. The 4-in. anemometer was calibrated for these tests by the Bureau of Standards and the new 3-in. instrument was calibrated by the manufacturer before delivery. The method of holding the instrument while taking a traverse on a small grille is shown in Fig. 8. The authors found that the 3-in. anemometer was more easily handled and is more applicable to the method of traverse to be recommended by this paper.

The following methods of traverse were employed:

1. *Spot traverse with straight average.* In this method the 12 x 12 grilles were divided into nine equal zones, and a half minute reading was recorded for each zone.

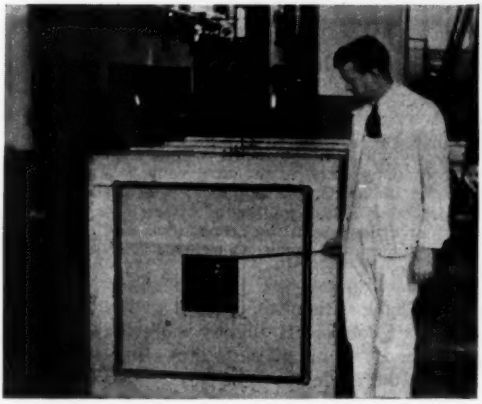

FIG. 8. APPLICATION OF ANEMOMETER

A straight average was obtained which was obviously too low, due to the fact that too much value was given the readings along the edges.

2. *Continuous traverse across the grille.* In this method the anemometer is moved back and forth across the grille without repeating on any strip. This method gives approximately the same results as the previous one. Methods 1 and 2 were recommended by Davies.[6]

3. *Zone traverse with weighted average.* The grille was divided into three horizontal zones and a moving traverse was made in each zone for a period of 1 min. In averaging these three readings the middle zone is given four times the value of the top and bottom zones. This is in accordance with Simpson's Rule in which the mean height of a parabolic curve may be determined by dividing the length into two equal parts, and then measuring the height of the three boundary lines of these parts. The mean height of the curve is the weighted average of four times the center line plus the height of each of the other two outer lines.

4. *Spot traverse with weighted average.* In this method the values of the nine

[6] The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies. (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 382.)

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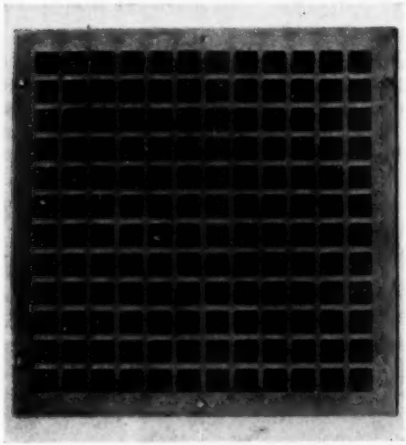

FIG. 9. TYPE 1 GRILLE 58.7 PER CENT FREE AREA

individual readings were weighted according to Simpson's Rule. In this case the center reading is multiplied by sixteen, each of the intermediate side values by four, and the corner values by one. Although this method is more nearly correct, it is too involved and time consuming to be of practical value.

5. *Moving traverse with automatic weighted value.* The grille was divided into three zones as in Method 3, and a continuous traverse was made with one crossing on the upper zone, four crossings on the center zone, and one crossing on the lower zone. This method automatically applied Simpson's Rule and gave the same results as methods three and four.

The authors applied the fifth method to all grilles and all sizes and found that it gave a curve of coefficients slightly more uniform than that from the straight average method. In all of these the anemometer was moved in contact with the face of the grille.

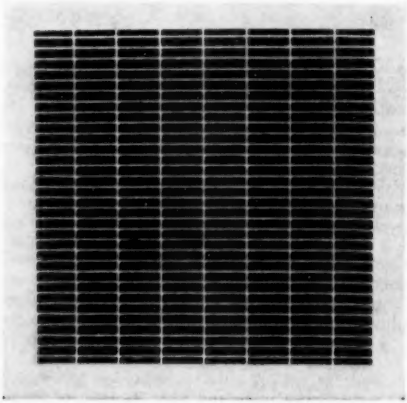

FIG. 10. TYPE 2 GRILLE 74.98 PER CENT FREE AREA

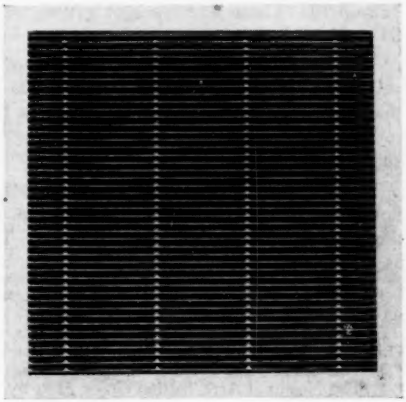

FIG. 11. TYPE 3 GRILLE 73.36 PER CENT FREE AREA

THE EXHAUST GRILLE COEFFICIENT *K*

Grille traverses were made in accordance with Method 5, which is termed: *Moving traverse with automatic weighted value.* Two or more runs were taken with each anemometer at each velocity, and the values were corrected by the calibration curves and averaged. The indicated volume of air passing the grille is, of course, obtained in accordance with the Equation of Continuity of Flow by multiplying the indicated velocity by an area, and, if necessary, a discharge coefficient. One of the major problems of this paper was to determine which grille area should be used, and, having chosen a particular area to find out how it applied to grilles of various sizes and various per cents of free area.

The type grilles used in this investigation are shown in Figs. 9, 10, 11,

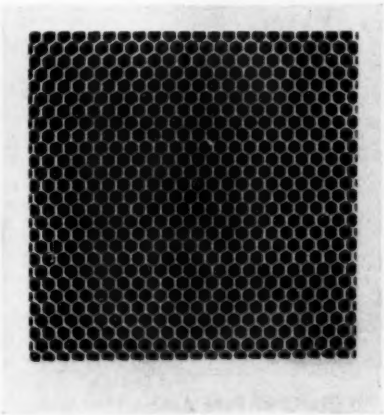

FIG. 12. TYPE 4 GRILLE 90.5 PER CENT FREE AREA

12 and 13. These five types of grilles were tested in the following four sizes: 24x6; 12x12; 24x12; 24x24.

This paper summarizes the results of tests on 20 grilles at anemometer velocities varying from 200 fpm to 1000 fpm. It will be noted in the curves that follow that the velocity range was limited on the largest size grille due to capacity limitations of the fan on account of the excessive drop in pressure at the nozzle.

The discharge coefficients of these grilles were found by dividing the true flow from the calibrated nozzle by the volume indicated by the product of the anemometer reading and an area. The following areas were considered as possibilities:

1. The Free Area, which is defined as the open area of the grille through which the air actually passes.
2. The Gross Frontal Area is the smallest area of the inside of the frame of the grille. This is the area which the anemometer crosses in making the traverse.
3. The Mean Area is the average of the Free Area and the Gross Frontal Area.
4. The Nominal Area (called Gross Area by Davies) is the area of the duct.

The per cent free areas as given in this paper are determined by the ratio:

$$\text{Per Cent Free Area} = \frac{100 \times \text{Free Area}}{\text{Gross Frontal Area.}}$$

All these values were computed by the authors and compared with those furnished by the manufacturers, although the authors' values were used in all cases.

Upon studying the 20 curves of coefficients plotted against velocity, the following observations are made:

1. The coefficient as determined from the gross frontal area, as in Fig. 14, is relatively constant over the complete range of grilles including all types and all sizes,

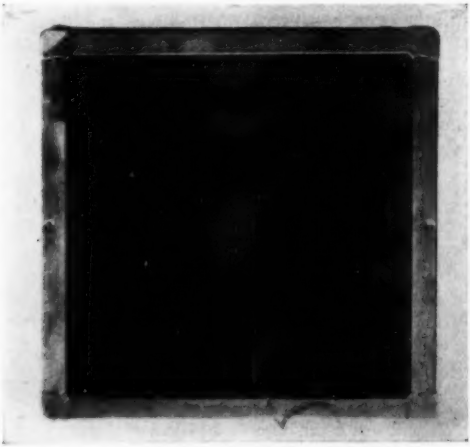

FIG. 13. TYPE 5 GRILLE 77.6 PER CENT FREE AREA

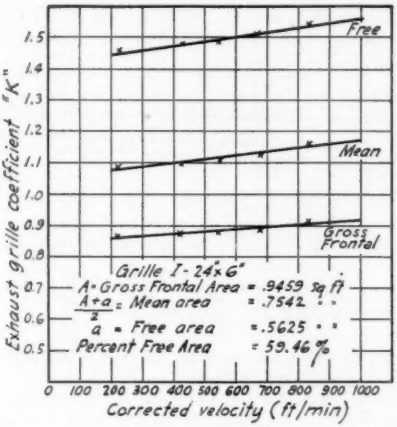

FIG. 14. COEFFICIENTS ON GROSS AND FREE AREAS

whereas the coefficients using the mean and free areas vary inversely as the increasing per cent free area. This definitely shows that the coefficient should not be based on the mean or free areas, unless the lower coefficient is to be used to illustrate the advantages of the grilles of higher per cent free area.

2. The curves, such as shown in Fig. 15, demonstrate the effect of size of grille and show that the smaller grilles have a slightly higher coefficient. A study of these curves and the hydraulic radii of the duct was used in developing the empirical formula to be given later.

3. Figs. 16 and 17 show a striking proof of the value of using the gross frontal area instead of the previously used nominal area. The gross frontal areas for the various types of grilles are given:

Grille No. 1 Gross Frontal of $12 \times 12 = 0.9604$ sq ft.
Grille No. 2 Gross Frontal of $12 \times 12 = 0.9538$ sq ft.
Grille No. 3 Gross Frontal of $12 \times 12 = 0.8633$ sq ft.
Grille No. 4 Gross Frontal of $12 \times 12 = 0.8556$ sq ft.
Grille No. 5 Gross Frontal of $12 \times 12 = 0.8636$ sq ft.

By examining the points on Fig. 16, which gives the exhaust grille coefficients based on the nominal or duct area, it is seen that Grilles 1 and 2, which have much larger gross frontal areas, lie above the average dotted line, whereas the remaining grilles lie below the line. Hence, by using the gross frontal area, the values are brought nearer to a straight line as shown in Fig. 17. The possible per cent of error is practically halved by using the curve as determined by the gross frontal area, because an allowance for the design variations of the various manufacturers has been made.

It is then proposed that the following equation and coefficients be used:

$$Cfm = K\,V\,A \qquad (7)$$

where

K = Values as given in the tabulation of the next paragraph.
V = Velocity in feet per minute as indicated by a calibrated anemometer using Method 5.
A = Gross frontal area in square feet.

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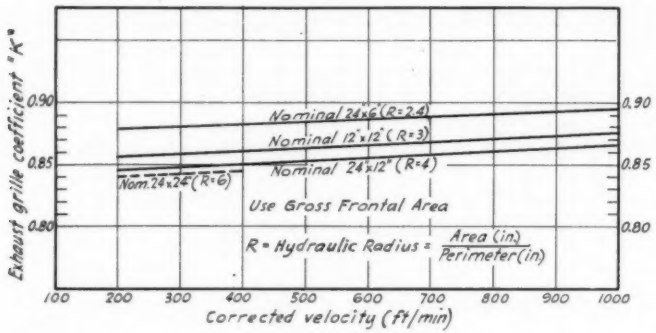

FIG. 15. CURVES SHOWING VARIATION OF MEAN COEFFICIENT WITH GRILLE SIZE

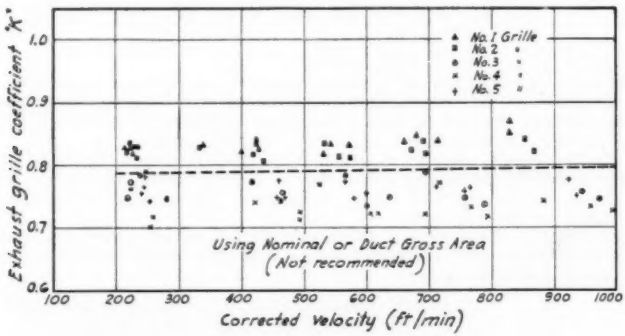

FIG. 16. COEFFICIENTS ON DUCT AREA

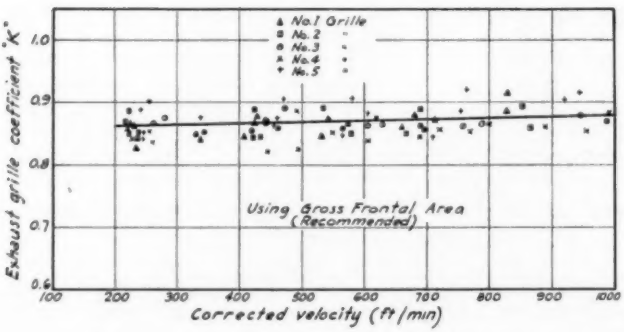

FIG. 17. COEFFICIENTS ON GROSS FRONTAL AREA

By using a value of $K = 0.873$ for all grilles and sizes, the maximum observed error was ± 6.07 per cent and the average error was ± 3 per cent.

The maximum error resulting from the use of a simpler coefficient was reduced to ± 5.83 per cent, by the use of coefficients varying with velocities. These values of K for different velocities were found from the curves and are given:

VELOCITY FPM	COEFFICIENT
200	0.863
300	0.865
400	0.867
500	0.871
600	0.872
700	0.875
800	0.877
900	0.880
1000	0.882

By using the accompanying coefficients for grilles of different sizes, the maximum recorded error was ± 4.62 per cent. These values were taken from diagrams of the investigation, using 600 fpm as the average velocity.

K of Nominal 24 × 6 grille = 0.886
K of Nominal 12 × 12 grille = 0.865
K of Nominal 24 × 12 grille = 0.855
K of Nominal 24 × 24 grille = 0.842 (Calculated)

The velocity effect on the given values of K may be cared for by multiplying each of the factors by

$$1 + \frac{V - 600}{40{,}000} \qquad (8)$$

and the resulting value of K showed a maximum error of ± 4 per cent.

For grilles of all sizes including those given, the effects of size and velocity may be included by using the formula in determining the value of K:

$$K = \frac{0.904}{R^{0.04}}\left(1 + \frac{V - 600}{40{,}000}\right) \qquad (9)$$

in which

K = Coefficient in formula $Cfm = K\,V\,A$

R = Hydraulic Radius = $\dfrac{\text{Nominal Area in square inches}}{\text{Nominal Perimeter in inches}}$

V = Velocity in feet per minute as indicated by the corrected anemometer using Method 5.

The maximum recorded error in using this formula was ± 5.65 per cent and the average was ± 3.22 per cent.

This remaining error is due to the variation in anemometer readings and to the frictional effect of the grille members. This latter error can only be eliminated by using the test grille coefficients which are recorded by most manufacturers for each of their grilles.

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CONCLUSIONS

This investigation has checked the work of Davies and established an improved method of determining the quantity of air flowing through a modern exhaust grille. It has been demonstrated that a relatively constant coefficient of $K = 0.873$ is obtained when the gross frontal area is used in the formula:

$$Cfm = K\,V\,A \qquad (10)$$

The average coefficient for all grilles and all sizes has been changed from the Davies' value of 0.80 to 0.873. These results were obtained by Method 5, *moving traverse with automatic weighted value,* as proposed in this paper.

The variation in K, the maximum recorded error of which was 6 per cent on each side of the mean curve, is dependent upon velocity, grille size, and design. The first two of these variables were cared for in the empirical formula derived from the observed data, whereas the authors were unable to find a law correcting for grille design. The empirical formula is:

$$K = \frac{0.904}{R^{0.04}}\left(1 + \frac{V - 600}{40{,}000}\right) \qquad (11)$$

When greater accuracy is desired, it is necessary to use coefficients determined for the particular type of grille form and hydraulic radius of duct.

APPENDIX NO. 1

Equation 5 for finding the velocity from the Pitot velocity head, $V = 4006\sqrt{h/x}$ is derived as follows:

$v^2 = 2gh$
$v = \sqrt{2gh}$

Volume of 1 lb of dry air at 70 F = 13.35 cu ft.
Weight of 1 cu ft water at 70 F = 62.295 lb.

$$\text{Velocity in feet per minute} = 60\sqrt{\frac{64.4 \times \frac{62.295 \times 13.35}{12} \times h}{x}}$$

$$\text{Velocity in feet per minute} = 4005.9\sqrt{h/x} = 4006\sqrt{h/x}$$

in which:

V = Velocity in feet per minute.
h = Velocity head in inches of water at 70 F.
x = Density of actual air relative to air at 70 F and 29.92 in. Hg.

APPENDIX NO. 2

Equation 6 in this paper is derived as follows: From thermodynamics and the principle of conservation of energy, the equation for the gain in kinetic energy of a perfect gas from the change in enthalpy (equation of flow) is

$$\frac{V_2^2 - V_1^2}{2g} = \frac{k p_1 v_1}{k - 1} - \frac{k p_2 v_2}{k - 1} \qquad (1)$$

this gives

$$\frac{V_2^2}{2g}\left[1-\left(\frac{V_1}{V_2}\right)^2\right]=\frac{k}{k-1}(p_1v_1-p_2v_2)$$

and

$$\frac{V_2^2}{2g}=\frac{\frac{k}{k-1}p_1v_1\left(1-\frac{p_2v_2}{p_1v_1}\right)}{1-\left(\frac{V_1}{V_2}\right)^2} \quad \ldots\ldots(2)$$

but

$$\frac{p_2v_2}{p_1v_1}=\frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{\frac{1}{k}}=\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \quad \ldots\ldots(3)$$

and

$$V_1=\frac{Vol_1}{A_1}=\frac{MRT_1}{p_1A_1} \qquad V_2=\frac{MRT_2}{p_2A_2}$$

hence

$$\frac{V_1}{V_2}=\frac{T_1A_2p_2}{T_2A_1p_1}$$

but

$$\frac{T_1}{T_2}=\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}$$

and therefore

$$\frac{V_1}{V_2}=\frac{A_2}{A_1}\left(\frac{p_2}{p_1}\right)^{\frac{1}{k}} \quad \ldots\ldots(4)$$

substituting equations (3) and (4) in equation (2):

$$\frac{V_2^2}{2g}=\frac{\frac{k}{k-1}p_1v_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}}$$

but

$$p_1v_1=RT_1$$

hence

$$\frac{V_2^2}{2g}=\frac{\frac{k}{k-1}RT_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}}$$

and

$$\text{Volume per second}=A_2V_2=A_2\sqrt{2g\frac{k}{k-1}R}\sqrt{\frac{T_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}}}$$

$$\text{Volume per second}=A_2\sqrt{64.4\,\frac{1.4}{0.4}\times 53.34}\sqrt{\frac{T_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}}}$$

$$=110A_2\sqrt{\frac{T_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}}} \quad \ldots\ldots(6)$$

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In order to simplify

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} = \left(\frac{p_1 - \Delta p}{p_1}\right)^{\frac{0.4}{1.4}} = \left(1 - \frac{\Delta p}{p_1}\right)^{\frac{0.4}{1.4}}$$

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} = 1 - \frac{0.4}{1.4}\,\frac{\Delta p}{p_1}$$

(since $\frac{\Delta p}{p_1}$ is of the order of $\frac{1}{50}$ the higher powers are neglected)

Similarly

$$\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} = 1 - \frac{2}{1.4}\,\frac{\Delta p}{p_1}$$

Substituting in equation (6)

$$\text{Volume per second} = 110A_2\sqrt{\frac{T_1\left[1 - \left(1 - \frac{0.4}{1.4}\,\frac{\Delta p}{p_1}\right)\right]}{1 - \left(\frac{A_2}{A_1}\right)^2\left(1 - \frac{2}{1.4}\,\frac{\Delta p}{p_1}\right)}}$$

$$\text{Volume per second} = 110A_2\sqrt{\frac{\frac{T_1}{p_1}\left(\frac{0.4\Delta p}{1.4}\right)}{1 - \left(\frac{A_2}{A_1}\right)^2\left(1 - \frac{2\Delta p}{1.4}\right)}}$$

but

$$\frac{T_1}{p_1} = \frac{v_1}{R}$$

$$\therefore \text{Volume per second} = 110A_2\sqrt{\frac{\frac{0.4}{1.4} \times \frac{1}{53.34} \times \Delta p\, v_1}{1 - \left(\frac{A_2}{A_1}\right)^2\left(1 - 1.4286\,\frac{\Delta p}{p_1}\right)}}$$

or

$$\text{Volume per second} = 8.62A_2\sqrt{\frac{\Delta p \times v_1}{1 - \left(\frac{A_2}{A_1}\right)^2\left(1 - 1.428\,\frac{\Delta p}{p_1}\right)}}$$

Note: This is volume at discharge. The volume at entrance is equal to the above multiplied by the factor $\left(1 - 0.714\,\frac{\Delta p}{p_1}\right)$.

No. 1093

THE CONDENSATION NUCLEI CONTENT OF THE AIR AS RELATED TO AIR FRESHNESS

By R. A. NIELSEN * *(NON-MEMBER)*, EAST PITTSBURGH, PA.

INTRODUCTION

THE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1938, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS states clearly the present status of air conditioning as related to air freshness:

"In spite of the rapid advances in the field of air conditioning during the past few years, the secret of reproducing indoor atmospheres of as stimulating qualities as those existing outdoors under ideal weather conditions has not as yet been found. Extensive studies have failed to elucidate the cause of the stimulating quality of country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical means."

This paper presents the results of some experiments which were performed in order to investigate the possible relationship between the freshness of air and its condensation nuclei content. Condensation nuclei are those things which, when saturated air is expanded, will serve as centers for the formation of water droplets. A great many measurements upon the number of such nuclei in the air have been made with a view to determining the relation between the variation in their number and certain atmospheric electrical phenomena. In mountain and ocean air the average number of condensation nuclei is a few hundred per cubic centimeter; in the country it is a few thousand; and in cities the concentration is much higher. A series of nuclei counts taken in the vicinity of Pittsburgh within a short time and on a cloudy day illustrates the variation in the concentration of condensation nuclei to be expected between city and country air as shown in Fig. 1. The counts in the city ranged from 75,000 to 700,000 nuclei per cubic centimeter depending upon the wind direction, the nearness to industrial plants, congested traffic areas, etc. Outside the city the nuclei count rapidly diminished and there was a marked decrease in the fluctuations between readings. Farther from the city the counts continued to decrease, till at 10 to 15 miles away, the counts consistently were very near 25,000 per cubic centimeter.

It has been reported [1] that the number of condensation nuclei in occupied

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[1] A Cause for the Decrease in the Number of Ions in Air of Occupied Rooms, by G. R. Wait (*Journal of Industrial Hygiene*, Vol. 16, No. 3, May, 1934, p. 147).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

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rooms increases with the time and the number of occupants. Since air staleness also increases with the number of occupants in a room, since mountain air has those qualities of freshness which are desired, and since the mountain air in contrast to city air is fresher and has a lower nuclei content, it appears that it might be possible to use the condensation nuclei content of the air as an index of its freshness.

An Aitken nuclei-counter, as shown in Fig. 2, is ordinarily used to count these nuclei. The instrument consists of a chamber of about 10 cubic centimeter volume with a small piston attached. The bottom of the instru-

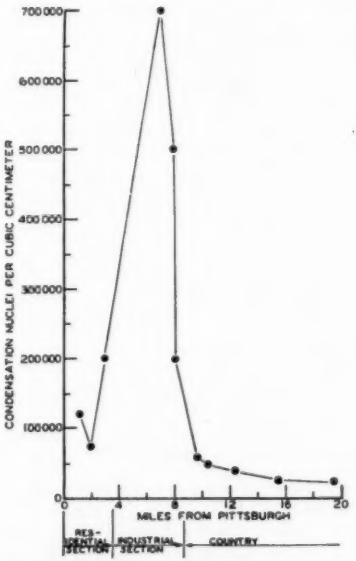

FIG. 1. CONDENSATION NUCLEI CONTENT OF THE AIR IN THE VICINITY OF PITTSBURGH

ment is a piece of ruled glass. Above the plain glass top of the chamber is a magnifying eyepiece focused on the ruled glass below; the sides of the chamber are covered with a damp material such as blotting paper. An air sample is introduced into the counter and allowed to become saturated; it is then quickly expanded about 25 per cent. This expansion produces sufficient super-saturation of the air to form a cloud of droplets which fall on the ruled glass surface where they can be counted. The air is then returned to its original volume and allowed to come to temperature equilibrium before being expanded again. Upon the second expansion, fewer but larger droplets form and fall. If the process is repeated three or four times, droplets cease to form, *i.e.*, all nuclei of condensation have been removed. From the number of droplets that have fallen on the ruled glass at each expansion, the number

of nuclei per cubic centimeter in the sampled air can be calculated.[2] Where the concentration of nuclei is large, it is customary to dilute the sampled air with nuclei-free air so that the number of droplets obtained is easily countable.

The nuclei count of laboratory or outside air was found to be only slightly dependent upon the technique used in making the counts. For example, whether the air sample was allowed to remain in the counting chamber 2 or 20 seconds before expanding it for the first time; whether the expansions were 15 or 25 per cent; whether they were slow or rapid, the counts obtained were substantially the same. Table 1 gives the results of a series of nuclei counts using laboratory air which in all cases was diluted with 50 volumes of

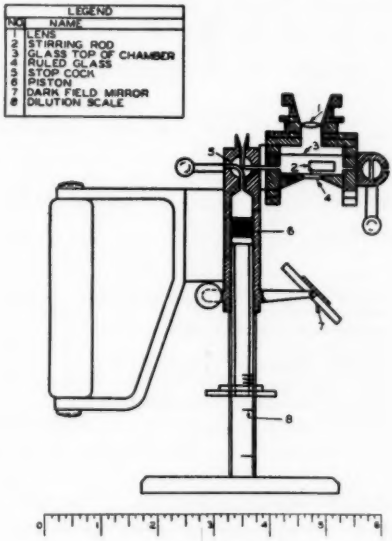

FIG. 2. NUCLEI-COUNTER

nuclei-free air. Example No. 1 gives the values of droplet counts obtained using eight different samples of air. The average value for this series of counts is 10⅜. Multiplying this by 5000 (100 times the dilution) gives 51,900 for the number of nuclei per cubic centimeter in the laboratory air.

Better results were obtained more easily when using the 25 per cent expansions since, in this case, the droplets were larger and fell more rapidly and evaporated more slowly; thus they remained visible longer and were easier to count. Nuclei in a closed chamber disappeared exponentially with time. The disappearance was accelerated by stirring since it was caused, in part, by the impinging and sticking of the nuclei on the surfaces of the counting chamber. The number of droplets formed was found to decrease with the length of time

[2] The Aitken Pocket Nuclei-Counter, by G. R. Wait (Gerland's *Beitrage zur Geophysik*, Vol. 37, 1932, p. 429).

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the sampled air was allowed to remain in the saturated atmosphere of the counting chamber. (See Nos. 1 and 2 in Table 1.) The number of droplets counted after the sample had remained in the counter chamber for 10 min was about half as many as would have been obtained if the count had been made within a very few seconds after the air sample was taken into the chamber. Counts obtained by expanding the sample immediately upon its entering the counter, fluctuated between wide limits since sufficient time was not allowed for mixing the sample with the nuclei-free air with which it was diluted. A small stirring rod, operated by tilting the counter, hastened the process of mixing so that within a very few seconds the count could be made and good results obtained. Nevertheless, quite wide variations could be expected between successive counts. An individual count may be somewhat unreliable, but the average of a series of counts was reproducible.

TABLE 1—RESULTS SERIES OF NUCLEI COUNTS USING LABORATORY AIR

No. 1	9–13–10–7–8–14–7–15	Avg. 51,900 nuclei per cubic centimeter
No. 2	7–9–10–5–8–8–11	Avg. 41,400 nuclei per cubic centimeter
No. 3	10–11–9–12–10	Avg. 52,000 nuclei per cubic centimeter
No. 4	11–11–10–8–10–8–13	Avg. 50,700 nuclei per cubic centimeter
No. 1	3 seconds elapsed between sample-taking and the first expansion	(Rapid 25 per cent expansion)
No. 2	20 seconds elapsed between sample-taking and the first expansion	(Rapid 25 per cent expansion)
No. 3	3 seconds elapsed between sample-taking and the first expansion	(Rapid 15 per cent expansion)
No. 4	3 seconds elapsed between sample-taking and the first expansion	(Slow 25 per cent expansion)

The condensation nuclei content of air can be reduced by passing the air very slowly through filters, or by allowing the air to remain for a long time in a closed container. Before discussing the effect on air freshness of reducing the number of condensation nuclei in the air, the characteristics of some nuclei which were produced when air was subjected to processes used in air conditioning will be described.

SOME CHARACTERISTICS OF CERTAIN NUCLEI

Although illuminating gas contained very few nuclei, air sampled from the vicinity of a gas flame contained hundreds of thousands of nuclei per cubic centimeter. These nuclei were probably both gaseous and particulate matter resulting from the burning of the gas. (That gases, as well as particles, can act as condensation nuclei can be easily illustrated by adding a little sulphur dioxide to the air.) Investigation of some irregularities in the counts revealed that if the expansions were made immediately upon taking the air samples, the counts obtained were from two to five times greater than those obtained when the air was allowed to remain in the saturated counting chamber a few seconds

before it was expanded. The sooner the initial expansion could be made, the greater was the number of nuclei counted.

It was found that when a surface was heated, nuclei were given off. These nuclei were many times more easily removed by a thin paper filter than were those which were normally in the air. As with the gas flame nuclei, the counts obtained depended upon the counting technique. Nuclei-free air was heated and, for one particular case, the counts under steady conditions were as follows:

Immediate expansion	10 droplets per square
10 second wait before expansion	4 droplets per square
30 second wait before expansion	1 droplet per square

In some cases if one waited 10 seconds between the taking of the sample and the expansion, the count obtained was only 1/25 of that obtained when the sample was expanded immediately upon drawing it into the counter.

To discover what these nuclei were, it was decided to try to collect them by freezing them out of the air. However, the chilling of a surface was also found to produce nuclei. The colder the surface, the greater was the rate of nuclei production. When nuclei-free air was passed through a cold glass tube, nuclei were first produced at temperatures a little below the freezing temperature of water. The rate of production increased rapidly with the decrease in temperature of the surface and was very great at liquid air temperatures. These nuclei acted like those in ordinary air; the value of the counts being much less affected by the counting technique than was the case in the two previous examples.

Air samples taken near a low pressure mercury arc (the type used as a source of ultra-violet light) were found to contain great quantities of condensation nuclei. An estimated value for their concentration was over a million per cubic centimeter. When rapid expansions were used, fine clouds of very slow-settling nuclei were obtained. The droplets often were so small that they were visible only under very good illumination; also, most of them evaporated before they fell on the ruled glass, where they could be counted. Many times fewer nuclei were obtained by using a slow instead of a rapid expansion.

Nuclei-free air was passed through a quartz tube. Immediatly upon being irradiated by the ultra-violet light from the mercury arc, nuclei were produced in great quantities. The number increased with the intensity of the light. After a few minutes, the rate of nuclei production decreased to a low but steady value. These nuclei disappeared rapidly with the length of time that they remained in the saturated counting chamber before the sample was expanded. It was estimated that the count dropped to ⅕ of its value by waiting five seconds between the taking and the expansion of the sample. Another 80 per cent decrease was observed for the next five seconds' waiting. A filter was found to be much more effective in removing these nuclei than in removing those normally found in the air.

A few of the peculiarities observed in counting different kinds of nuclei have been given in the previous discussion. These examples should be sufficient to illustrate the transient and peculiar nature of certain kinds of condensation nuclei. When dealing with classes of nuclei where slight changes in the technique of making the counts introduce into those counts variations of such large magnitude, the meaning of the counts of such nuclei must be questioned.

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It was not the purpose of this experiment to investigate the relation between different types of nuclei, different counting techniques, and air freshness; rather, it was the purpose to see if, when using that technique which is used in atmospheric electrical work and which gives reproducible counts in normal air, a correlation existed between the nuclei count and the air freshness.

CONDENSATION NUCLEI AND AIR FRESHNESS

A 200 cu ft all-metal room was equipped with air ducts to recirculate and treat the air. The air samples for the nuclei counts were taken from the outside through a 1/32-in. hole in one of the metal sidewalls. The nuclei count of the air in the metal room was reduced from 50,000 to only 700 per cubic centimeter by recirculating it through filters. The recirculation was then stopped and the room left closed; in one and one-half hours, the nuclei count had risen to 1000 per cubic centimeter. The increase was undoubtedly due to slight air leaks around the door of the room. One person entered the room and closed the door as quickly as possible, but in so doing, increased the count to 2000 per cubic centimeter. Half an hour later with one person inside, the count had increased to 6000 per cubic centimeter; the rise thereafter was somewhat slower. The air became very stale in the room, even though the nuclei count was low. Since the ventilating ducts were closed, a small fan within the room was used to circulate the air so that the person inside would be reasonably comfortable. When the occupant left the room (nuclei count 7000 per cubic centimeter), the laboratory air (nuclei count 40,000 per cubic centimeter) seemed very fresh by comparison. When one first entered the room with the low nuclei count, no noticeable feeling of freshness was experienced, such as was the case upon leaving that room after about an hour's occupancy. The test was repeated, but this time the nuclei count was gradually reduced, by recirculating the air through filters, to a concentration of only 1/10 of that in the outside room. Nevertheless the air within the room became oppressive. If one replaced the paper filters with charcoal ones, the air quality was improved, but it was, nevertheless, refreshing to leave the metal room and to again breathe regular laboratory air.

The nuclei count in the metal room was reduced by leaving it closed over the week end. At the start of the experiment, the nuclei count was 30,000 per cubic centimeter. Two and one-half days later the nuclei content was only 2400. Nevertheless when one entered this room of low nuclei count, it was found that the air lacked freshness.

On another occasion it was found that on one *smoggy* morning the nuclei count outside the laboratory window was about 200,000 per cubic centimeter while inside it was 75,000. In this case, there was no noticeable increase in freshness associated with the inside air. A series of nuceli counts, made in the country at the side of the highway when the traffic was light or when there was a wind to blow the exhaust products of the automobiles away from the nuclei-counter, gave consistent values of a few thousand nuclei per cubic centimeter; occasionally a count of about 100,000 nuclei per cubic centimeter was obtained. A minute before or after this high count, however, the values were normal. No noticeable change in air freshness accompanied these sudden but great changes in nuclei content of the air.

The increase in the concentration of nuclei in the air of the metal room due to occupancy by one person agreed well with the values expected from previous experiments.[3] However, from the results of the present experiments, no verification of the statement that the breath contained nuclei generated in the lungs can be given. Air from the breath was sampled directly and was found to contain fewer nuclei than the inhaled air. When a balloon was distended by many short breaths, there were more nuclei than when it was distended by the second half of a deep breath. Air sampled from the second half of the breath contained from 2000 to 5000 nuclei per cubic centimeter; the inhaled air in this case contained 50,000. If the air were sampled from the first part of the breath, *i.e.*, air that had been in the mouth and throat, values between 5000 and 40,000 were obtained. In no case was the nuclei count of the breath greater than that of the inhaled air. Even in the metal room where the nuclei count was low, whether the air was blown directly into the intake of the nuclei-counter, or into the vicinity of the counter, or whether the air sample was drawn into the counter slowly or rapidly, the nuclei counts so obtained were never larger than the count of the inhaled air, even when this was as low as 2000 per cubic centimeter.

CONCLUSIONS

Air lacks freshness when industrial and other processes fill it with high concentrations of condensation nuclei; however, these experiments indicate that the reduction of the nuclei content of air from the high concentrations found in the city to low concentrations, equivalent to those of country or mountain air, adds nothing to its freshness. Since processes used in air conditioning produce nuclei having such transient and erratic behavior in a nuclei counter, it appears that the concentration of these nuclei is not a usable index of the freshness of the air in conditioned atmospheres.

DISCUSSION

W. A. DANIELSON (WRITTEN): This paper indicates that the search for the cause of *freshness* in air must follow other trails, that may in the end be blind. Since oxygen and CO_2 content evidently have little, if any, bearing on this freshness and with deodorizing filters in the offing, the operating cost of winter heating and summer cooling, with walls and roofs of low heat conductivity, will approach the limits of the average man's pocket book. If then the invigorating effect can be produced at low cost, a market will be created that will help the depressing recession. The search for the facts relative to this should be directed in every way that offers any possibility of success.

One condition that this paper on condensation nuclei does not take into consideration is that, if the Brownian movement of the charge of air is of the same polarity as the walls of the chamber, then the discharge of the nuclei would undoubtedly be delayed. If the two elements were of different polarity, the opposite effect would undoubtedly be the case. The high humidity would accelerate the neutralization of the electric charges, but the time factor is so short that this may not completely take place.

This study causes one to consider the statement so often made that each rain drop was originally formed around a particle of dust. Whether this is true or whether

[3] Loc. Cit. See Note 1.

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the particles of ever present dust just happen to join the droplet, it is believed has not definitely been proved. It would be interesting to clean a quantity of air thoroughly, then saturate it with clean water vapor, and try for condensation of nuclei. The electrical conductivity of the water vapor would then be the only probable variable, and that might finally be made uniform so that the changing of the water particle's action would result in similar Brownian movements. This tremendous force in nature will undoubtedly be found to continue a disturbing element in research of this character.

DR. S. L. WARREN[4] (WRITTEN): The experimental data reported by R. A. Nielsen are extremely interesting. In reviewing the subject under discussion, I have been much confused in my attempt to determine a good definition for fresh air. In setting up a definition for comfort, it is obvious that the air should be as free of dust as possible and that the humidity and temperature and oxygen and carbon dioxide ratio should be within certain limits. Yet these factors are not of themselves enough to bring about that exhilaration which we term freshness in the air. A good deal of the latter, of course, is a subjective phenomenon and is determined by the odor, the contrasting temperature and humidity, a lack of obvious dust particles, and so on. Frequently in passing from one room to another or to the out-doors one is impressed by the freshness, chiefly because one passed from a high humidity, high temperature environment into a lower humidity temperature relationship, with increased evaporation of the perspiration and a perceptible change in the skin temperature and the feeling in the nasopharyngeal passages of a reduced temperature and a new or more enticing odor than was present in the environment which was just left.

It is a common experience to notice that after remaining in such a stimulating environment the stimulation gradually subsides and the freshness disappears. In a room fouled by human occupancy there are not only changes in the humidity and temperature and oxygen—carbon dioxide ratio, but there are also many body odors which are given off by the sweat glands and to a certain extent ammonia and other volatile substances coming from the skin and from the lungs which have a depressing influence upon one breathing such air. All these together with dust particles tend to destroy the freshness. There is no doubt that a good many of these substances contribute to the high ion counts in rooms. It seems that this problem should be investigated from the subjective standpoint as well as from the purely physical aspects.

L. R. KOLLER[5] (WRITTEN): This paper presents a valuable investigation of the relation, if any, between air freshness and the number of condensation nuclei present. The writer finds that the concentration of nuclei is not a usable index of the freshness of the air in conditioned atmospheres.

The experiments performed in the all-metal room would have been more convincing if the temperature and humidity in the room could have been maintained the same as outside in the laboratory. It seems probable that the stuffiness in the metal room, due to high humidity and temperature as well as body odors, would far outweigh any possible effect due to the difference in nuclei count between this room and the laboratory. The experiment in which the count in the metal room was reduced by leaving it shut over the weekend is not open to this same criticism.

From Mr. Nielsen's measurements made on outdoor air, it is evident that nuclei content alone cannot account for freshness or lack of freshness of air. It seems that the factors of temperature, humidity and the content of foreign gases or vapors are probably more important.

DR. C. A. MILLS (WRITTEN): This paper quite clearly tells its own negative story as to the importance of condensation nuclei in determining air freshness and little discussion is needed. We are still entirely in the dark about what freshness is, or upon what it depends. We can readily detect it or its absence in the air we breathe,

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[5] Research Laboratory, General Electric Co., Schenectady, N. Y.

but we have no means of its mechanical standardization. It seems more likely a condition of the air itself rather than a matter of nuclear content. Of methods so far offered, ozonation or activation of the oxygen probably comes nearest to producin the freshness principle.

Here lies one of the major functions of air conditioning, about which no usefully complete knowledge is yet available. It is essentially a problem for biologic research, since freshness is of interest only in its biologic effects. It would seem to me much more important that the Society through its Research Committee sponsor investigations along this and other fundamental lines, than that so much of its efforts should go into the details of the physical handling of air. Radiational control of body heat loss, both for winter heating and summer cooling, deserves your close attention, as well as this matter of air freshness. You are not air conditioning engineers exclusively I hope. Rather are you heating and ventilating engineers, whose interest should be to do the job in the best possible way. In recent years air conditioning has offered that best way. But now you should be seeking ways of doing a better and less costly job than is offered by air conditioning.

W. L. FLEISHER (WRITTEN): My general criticism of this paper is that it is completely negative in its result and consequently from the angle of air conditioning it can be put in the same classification as ionization for the time being. It would have been interesting if certain other information had been included in this paper such as the pressures used in the nuclei-counter and pressure after expansion as this might have answered the question of whether the results could have been altered by different methods of counting or application of pressures. It appears to me that this paper deals too much in generalization. For example, without known values of magnification it seems to me that this technique of measurement is far too inadequate to determine the size of particle for fog. The mention of the fact that ordinary filters remove the possibility toward formation of condensation nuclei seems inconsistent based on my own personal ideas that the reason for cloud formations develops around nuclei which are referred to in this paper. A particle around which a fog forms could not be removed by the filter referred to by Mr. Nielsen.

This paper has started an investigation of a subject which may be of tremendous importance outside of the field to which it specifically refers or to which the author or reader might imagine.

If this investigation leads to a method of eliminating small fog particles it would serve as a positive rather than a negative purpose. In other words I would like to see the method orientated to the electrical precipitator and whether the droplets referred to have charges or are polarized so that more effective dust elimination may result with different electric potentials in an expanded media.

W. D. FLEMING[6] (WRITTEN): It has been known that the number of condensation nuclei in a given volume of air is much higher in cities than in the country and that the number of these nuclei in a room increases with occupancy. The present paper reports an investigation of a possible correlation between the number of condensation nuclei in air and the freshness of this air.

An Aitken nuclei-counter was used. While large variations of count may be caused by slight variations of technique, the technique can be readily standardized to give reproducible results under given conditions and to show significant differences between the nuclei content of air in different situations.

The nuclei count was found to be increased by flames, heated surfaces, surfaces chilled below freezing and ultra-violet radiation.

In testing the relation of nuclei to air freshness a 200-cu ft all-metal room was used. The decrease of the nuclei in this room was obtained by circulating the air through filters. Even when the nuclei count in the room was much lower than in the laboratory outside, the air impression in the room was that of staleness. Air alongside a country road while found to be usually low in nuclei, showed momentary rises to

[6] Lieutenant Colonel, Army Medical School, Army Medical Center, Washington, D. C.

high levels. No noticeable change in air freshness accompanied these changes in nuclei content.

The author's conclusion that while a stale air has a high content of condensation nuclei, reducing this nuclei does not add anything to the freshness of the air, appears well proven.

G. R. WAIT[7] (WRITTEN): Authorities are now generally agreed that condensation nuclei found in the open air, especially over land-areas, consist of minute particles arising from combustion processes. In coastal regions small particles of salt originating from the ocean water make up an appreciable portion of the nuclei present. In a general way, air in the country districts contains fewer condensation nuclei than the air in a city, and in a mountainous district still fewer nuclei are present. In a similar way, the mountainous air is generally recognized as freshest, then the open country air and then the city air. We are thus led to suspect a relationship between the absence of condensation nuclei and the freshness of air. This relationship may however be only apparent and not direct. The results obtained by the author indicate that any relationship may be only apparent. His results also suggest that the absolute concentration of nuclei is not a proper index of the freshness of the air of an occupied room. The writer is inclined to believe on the other hand that the relative, rather than the absolute concentration of nuclei, may have considerable merit as an indication of the freshness of the air of an occupied room, and would like to see this matter tested.

This paper points out that the number of condensation nuclei in the air of a closed room increased with occupancy. These results agree with those of previous papers,[8] and are consistent with the interpretation that condensation nuclei are given off by human beings, probably largely from the lungs. The author could not verify this conclusion, however, by counting nuclei directly from the breath. This was to have been expected, since it has been found that condensation nuclei freshly from the breath are not easily counted with a nuclei-counter.[9] The tentative explanation given was that the nuclei freshly given off by human beings are surrounded with moisture, and while in this condition many adhere to the walls of the narrow passages and never arrive at the counting chamber of the instrument. This accounts for the two opposing results obtained previously[8] and verified by the author: *First*, that condensation-nuclei in a closed room undergo an increase in concentration with occupancy, due to the human system giving off nuclei, probably largely from the lungs; and *second*, that not a large number of nuclei are counted in air taken directly from the lungs.

Granting then that condensation nuclei are given off by the human system, it becomes important to determine if the presence of these particular nuclei in any way affects the freshness of air in closed occupied rooms. Mr. Nielsen states that upon emerging from the small 200-cu ft chamber into the large room, the air seemed fresher. It seems very reasonable to conclude that, in the large room, the proportion of nuclei of human origin was too small to noticeably affect the freshness of the air of that room, whereas in the small chamber the nuclei after 30 min or more of occupancy may have been mostly of human origin and that these may have been responsible for the lack of freshness of the air. In other words, these results might reasonably be interpreted as showing that the accumulation of nuclei of human origin does affect the freshness of the air of a closed room. These results however are by no means conclusive, but like the results of previous investigators point to the need for further carefully controlled investigational work. It is to be hoped that Mr. Nielsen may find the opportunity of going into these matters at greater length.

[7] Dept. of Terrestrial Magnetism, Carnegie Institution of Washington, Washington, D. C.
[8] Loc. Cit. See Note 1.
[9] Large-Ion and the Small-Ion Content of Air in Occupied Rooms, by G. R. Wait and O. W. Torreson (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 119).

No. 1094

AIR FILTER PERFORMANCE AS AFFECTED BY KIND OF DUST, RATE OF DUST FEED AND AIR VELOCITY THROUGH FILTER

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station, University of Minnesota.

IT IS difficult to set up a laboratory test procedure which will measure the characteristics of an air filter and give reliable data by which its performance on a particular type of work can be predicted. This is due partly to the lack of a simple, standard method of measuring the amount and quality of dust in the air, and partly to the difficulty of selecting test conditions which will simulate practical requirements for a filter.

In 1933 the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS adopted a Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. This code provides a method of performing an accelerated service test on a filter. A dust mixture prepared to represent the average dust found in air is mixed into the air stream leading to the filter. The amount of this dust in the air is measured before and after passing through the filter. Air velocity and limiting pressure drop across the filter are selected to meet average filter requirements.

In formulating this code there were many variables to consider and part of the conditions had to be selected arbitrarily to make up a workable test procedure. Some of the controversial questions were:

1. A suitable method of measuring the dust concentration in the air entering and leaving the filter.
2. The preparation of a synthetic dust mixture which would have the properties of a dust found in average air.
3. The maximum allowable dust concentration that could be used for the air entering the filter without handicapping the performance of the filter.
4. A representative air velocity through the filter during the test period.

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METHODS OF MEASURING DUST CONCENTRATION IN AIR

An accurate method of measuring the amount of dust in the air before and after passing the filter is an essential requirement of a test code. Several methods have been proposed, but no single method gives all of the properties of the dust which should be known for a complete analysis. Of these various proposals, the weight method was finally selected as the best adapted to an accelerated laboratory test. In using this method a known weight of dust is mixed with the air entering the filter, and the weight in the air stream leaving the filter is measured by passing a small percentage of this air stream through a fine grain alundum crucible. With careful laboratory technique this method gives accurate results, but it does not give any information as to dust particle size nor is it a practical method to be used outside of the laboratory or for ordinary dust concentrations in the air. It is a decided handicap to confine all filter tests to tedious laboratory procedure, and it would be very desirable if some other method could be devised for measuring dust concentration which would be practical as a field test and which might be substituted for, or else be used parallel with, the present code method.

SYNTHETIC DUST

The standard synthetic dust selected for the A.S.H.V.E. Air Filter Code consists of 50 per cent by weight of powdered lampblack containing a minimum of 97.5 per cent free carbon and having a minimum bulk value of 3.5 lb per cubic foot, mixed with 50 per cent by weight of Pocahontas bituminous coal ash screened to pass a 200 mesh screen. There are many varieties of dust in the air. The density, particle shape and size, as well as other characteristics, vary through wide ranges. Experience has shown that the type of dust passed through a filter may have a marked effect on its arrestance, and that a given dust may not have a like effect on the performance of two different types of filters. It is generally agreed that the dust in average air contains far less than 50 per cent by weight of carbon black. From this it is evident that the synthetic dust mixture as specified by the Code should have further investigation, and should be modified to simulate more nearly the dust which must be handled by the average filter.

DUST CONCENTRATION IN THE AIR

The Code specifies a dust concentration of 0.35 g (gram) of the mixture per 1000 cubic feet of air passing through the filter. This represents a much greater weight of dust per unit volume of air than would be found under ordinary conditions, but it is necessary to increase the weight in order to shorten the time of a filter test, and also to have a dust concentration in the air leaving the filters which can be cleaned out of the air and weighed by practical methods. The criticism of this part of the Code has been due to the uncertainty as to the dust arrestance of some types of filters when using heavy concentrations of dust as compared with the arrestance of these filters under normal operating conditions in which very much lighter dust concentrations would be used.

It has been contended that when heavy dust concentrations were used the viscous fluid on certain types of filters would not penetrate the initial layers

of dust rapidly enough to make them efficient in catching the succeeding dust particles, whereas under ordinary operating conditions more time would be available for the dust particles to be penetrated by the viscous fluid, and thus a continuously active dust-catching surface would be provided. This is a question which can be settled only by more experimental work.

AIR VELOCITY THROUGH FILTER

The optimum air velocity is probably different for different types of filters. For stationary filters the code specifies from 250 to 350 fpm through the gross face area of the filter, depending upon the type. There is a tendency in some types of apparatus to increase these velocities on account of the limited area available for installation of the filters. The increased velocity has a direct effect on the air pressure drop across the filter, and may have an effect on the arrestance of the filter. The effect on arrestance cannot, however, be predicted with the same certainty as the effect on air pressure drop.

TESTS ON RAILROAD AIR FILTERS

In the spring of 1937 the Research Committee of the *Association of American Railroads* decided to make a series of tests on those filters which were in common use for air conditioning work in railroad cars. Preparatory to these tests representatives of all filter manufacturers interested in railroad air filter work were invited to a conference to discuss the advisability of making certain changes in the test procedure as outlined by the Code. As a result of this conference, together with the results from an examination made of several samples of dust collected from railway air conditioning systems, a new dust mixture was selected and other slight modifications were made in the test procedure.

The dust mixture selected consisted of 50 per cent by weight of Pocahontas ash screened through a 200 mesh screen, 20 per cent by weight of Illinois fly ash screened through a 200 mesh screen, 20 per cent by weight of lampblack meeting the Code specifications, and 10 per cent by weight of Fuller's earth. These percentages were mixed together and screened through a 100 mesh screen to insure uniformity of mixture. The dust mixture was fed into the entering air at the rate of 0.4 g per 1000 cu ft of air passing through the filter, and the rate through the filter was set at 300 fpm face velocity. Thus the greatest deviation from the Standard Test Code was in the dust mixture selected. Later a test was devised for rating a filter on its capacity to remove lint from the air. This test, however, has no relation to the present investigation.

EXTENT OF PRESENT INVESTIGATION

The many questions which have arisen in regard to the adequacy of the Standard Test Code for determining the operating characteristics of different types of filters have shown the need for an extended investigation covering several phases of the subject. The present investigation covers three major points:

1. The relation between dust concentration in the air and filter arrestance.
2. The relation between air velocities through the filter and filter arrestance.
3. The relation of different types of dust and dust mixtures to filter arrestance.

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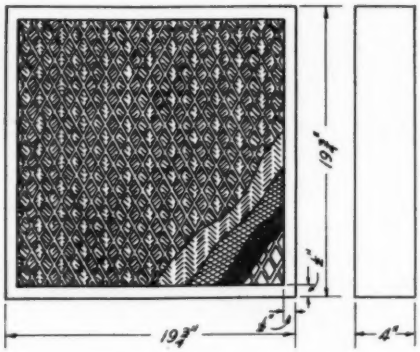
In making the investigation four commercial filters were selected as representative of four types in common use. These four types of filters were used throughout each series of tests and designated as Filters *A*, *B*, *C*, and *D*.

Filter *A*, shown in Fig. 1, is a recleanable type made up of 24 layers of metal screen as indicated in the sketch. These screens were graded in mesh and arranged to give a progressive pack for the filters. The filter was recondi-



Pack Arrangement	
Screen No.	Construction
1	Large Mesh Retaining Screen
2-5	Expanded Metal-Corrugated for Depth
6-11	Expanded Metal
12	Same as 6-11 but with three horizontal ribs ¼" in depth
13,15,17, 19,21,23	2 Layers copper mesh screen - 60 openings per sq.in.
14,16,18, 20,22	Same as 6-11
24	Rear Retaining Screen

FIG. 1. CONSTRUCTION DETAILS OF FILTER *A*

tioned between tests by thoroughly cleaning with trisodium phosphate, thoroughly drying, and dipping in a bath of filter oil at 150 F. The filter oil was allowed to drain for about 4 hours in an oven at 150 F. The properties of the filter oil are shown in Table 1. In order to expedite the work two of these filters were selected which had the same initial air resistance across the filter when cleaned but not oiled. After the filters were prepared for tests the air resistance across the filter was substantially the same. The slight differences were probably due to the difficulty of getting the filter absolutely clean after each test and covering it with a uniform film of oil. The differences in initial

pressure were very slight and were considered to have no major influence on the test results.

Filter *B*, shown in Fig. 2, is a throw-away type made from viscous coated fibers graded as indicated in the sectional view. New filters were selected for each test.

Filter *C*, shown in Fig. 3, was a throw-away type made up of two layers of cells set at an angle to each other such that the air in passing through the filter must impinge on the sides of the cell. The size of the cells was graded to give better distribution of the cleaning effect. New filters were used for each test.

Filter *D*, shown in Fig. 4, was made up of a light cotton filter pad glazed on one side. The filter material appeared to be very uniform in texture and the assembled filter gave substantially the same initial resistance for each test. In

TABLE 1—PROPERTIES OF VISCOUS COATINGS USED ON FILTERS *A*, *B*, AND *C*

OIL	FLASH POINT	FIRE POINT	POUR POINT	SAYBOLT UNIVERSAL VISCOSITY, SEC				PER CENT ASH
				70 F	100 F	130 F	210 F	
Filter *A*	505 F	575 F	10 F	4011	1222	486	93	1.50
Filter *B*	455 F	585 F	20 F	1170	426	230	...	2.64
					FUROL TUBE			
					100 F	122 F	210 F	
Filter *C*	560 F	600 F	60 F	...	2040	805	71	6.06

this case one frame was used for all tests, the filter media being replaced between tests.

There was naturally some slight difference in the initial resistance of the various filters selected from each group. These differences were small and were considered to have no material effect on the final results of the tests.

Filters *A*, *B*, and *C* were of the viscous coated type and the properties of the coatings as determined in the laboratory are shown in Table 1.

CHARACTERISTICS OF DUST USED IN TESTS

Some of the properties of the dust used in the tests are shown in Table 2. The densities were determined under two conditions, first loose, and second jolted to maximum density. For the loose densities, measurements were made by pouring the dust into a container without jolting and for the maximum density the dust in the container was jolted until no further settling took place. As may be noted from the figures in the table, the ratio between maximum density and loose density varies from 1.22 for lampblack to 1.68 for Pocahontas ash, with an average of about 1.5 for all types used.

The density of the mixed dust as used in the tests was less than the weighted densities of the constituent parts. For instance, the maximum density of a mix-

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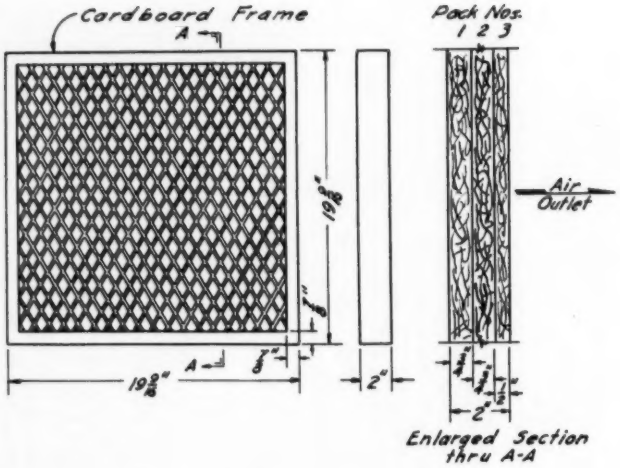
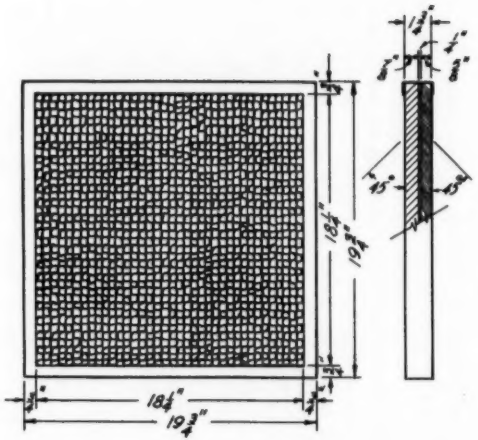


Pack No.	Thickness	Fiber Diameter	Fiber Length	Condition
1	¾"	252 Microns	12.25 ft./cu.in.	Oiled
2	¾"	252 Microns	12.7 ft./cu.in.	Heavily Oiled
3	¼"	135 Microns	14.2 ft./cu.in.	Oiled

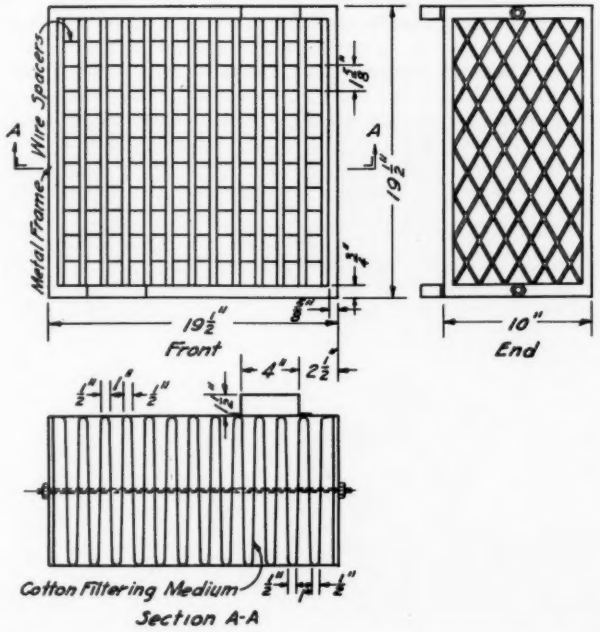
FIG. 2. CONSTRUCTION DETAILS OF FILTER *B*



Description	Pass No.1	Pass No.2
No. of cells per sq.in of Face Area	7.4	25.7
Width of Cell measured along Face in inches.	0.32 Ave.	0.16 Ave.
Length of Cell measured along Face in inches.	0.43 Ave.	0.24 Ave.

FIG. 3. CONSTRUCTION DETAILS OF FILTER *C*

ture consisting of 50 per cent Pocahontas ash, 20 per cent lampblack, 20 per cent Illinois fly ash, and 10 per cent Fuller's earth gives a maximum density of 0.480 g per cubic centimeter, or 23 per cent less than the weighted density of the constituent parts, which was 0.622 g per cubic centimeter. The maximum



Filtering Medium:-	Cotton - glazed on back side
Filtering Area:-	34.15 sq.ft.
Weight of Filtering Medium:-	9.85 grams per sq.ft.
Ratio of Area of Filtering Medium to Cross Sectional Area of Duct:-	12.3

FIG. 4. CONSTRUCTION DETAILS OF FILTER *D*

density of a 50 per cent mixture of Pocahontas ash and lampblack was 0.316, or about 31 per cent less than the maximum weighted density of the constituent parts, which was 0.461 g per cubic centimeter. All of the different dusts were passed through the screen as indicated for the tests, and in addition the fly ash and Pocahontas ash were screened through a 325 mesh screen for purpose of

analysis only. This screen passed 77 per cent of the fly ash and 75 per cent to 87 per cent of the Pocahontas ash. A 325 mesh screen will permit passage of particles having a minimum diameter in one direction of 43 microns. There was a considerable amount of variation in the results of this test on various samples of dust. As indicated in the last two columns, Fuller's earth showed the greatest moisture absorption with lampblack, and Cottrell ash next, and with Illinois fly ash and Pocahontas ash showing the least.

TABLE 2—PROPERTIES OF DUSTS USED IN TESTS

DUST	DENSITY GRAMS PER CC		SCREEN ANAL. PER CENT BY WEIGHT		MOISTURE LOSS PER CENT HEATED TO 175 F FOR 18 HOURS
	Loose	Jolted to Max. Density	Between 200 and 325 Mesh	Pass 325 Mesh	
Fuller's Earth Passed 100 mesh	0.412	0.600	...	...	5.1
Fly Ash from Ill. Coal Passed 200 mesh	0.488	0.740	Approx. 23	Approx. 77	0.2
Pocahontas Ash Passed 200 mesh	0.455	0.766	Approx. 19	Approx. 81	0.15
Cottrell Ash Passed 100 mesh	0.888	1.285	...	...	0.98
Lampblack Passed 100 mesh	0.128	0.156	...	...	1.18
50 Per Cent Pocahontas 50 Per Cent Lampblack	0.215	0.316	...	...	...
50 Per Cent Pocahontas Ash 20 Per Cent Ill. Fly Ash 20 Per Cent Lampblack 10 Per Cent Fuller's Earth	0.317	0.480	...	...	...

TEST APPARATUS AND PROCEDURE

With the exception of the dust feeding device the assembled test apparatus shown in Fig. 5 is the same as that specified in the A.S.H.V.E. Standard Code for Testing Air Cleaning Devices. The main test duct is 20 in. square and 9 ft long, connected by a reducing section 3 ft long and a 12-in. diameter section 5 ft long to an exhaust fan. The test filter is placed in the 20-in. square duct with the exhaust face about 8 in. from the reducing section. A direct current drive variable speed fan is used to draw the air through the filter, the volume being measured by a standard rounded orifice and the pressure drop across the filter being measured by a standard inclined water gage.

The dust feeding apparatus is shown at the left of Fig. 5 and consists of a revolving disc upon which the dust sample has been evenly distributed in a ring of uniform thickness. The disc containing the dust sample is placed on a rotating plate and a continuous uniform ribbon of dust is shaved from the out-

side edge of the dust ring by a specially designed knife edge fastened to the end of an air tube. Air is drawn through this tube by a low pressure air line and Venturi tube. The velocity of the entering air picks the ribbon of dust from the rotating plate and carries it through a tube to the distributing nozzle at the entrance of the filter test apparatus. The circular plate carrying the disc is rotated by a friction wheel in contact with the disc directly under the dust pick-up tube. The rotating plate is moved forward by a screw mechanism which is operated at a constant relative speed to the rotating plate. Thus the ring of dust moves up toward the cutting knife at a constant speed, and since

FIG. 5. ASSEMBLED VIEW OF TEST APPARATUS

the tangential velocity of the disc under the cutting knife is maintained constant by the friction wheel underneath, the dust ribbon is of uniform cross-section and there is a constant rate of dust feed to the filter.

In preparing the dust samples, aluminum discs similar in shape to phonograph records are used and placed in the dust distributing apparatus shown in Fig. 6. The disc is clamped down to a surface plate by an inner and outer ring. The required weight of dust is placed between these rings and leveled off by a rotating scraper or leveling device. With thin dust samples of less than about 60 g total weight it was necessary to use a vibrating apparatus in order to distribute the dust evenly over the plate. With thick samples of dust it is possible to omit this vibrating apparatus and still get uniform dust distribution.

In making a test the correct weight of dust for a one-hour period is distributed on an aluminum disc. This disc is placed in the distributing apparatus,

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the rate of feed being determined by the speed of the disc. The volume of air passing through the filter is adjusted by the speed of the fan or by controlling the fan discharge outlet, and the air pressure drop across the filter is measured at intervals by a water gage. The amount of dust in the air leaving the filter is measured by drawing a certain percentage of this air through a fine grain alundum crucible for the continuous period of test and noting the gain in weight of the crucible due to the dust separated from the air. The arrestance

FIG. 6. APPARATUS FOR DISTRIBUTION OF DUST SAMPLE ON PLATE

calculations are based on the results of a one-hour test. The dust arrestance for a one-hour period is calculated as given:

$$\frac{\text{Weight of dust fed to filter} - \text{Weight of dust in air leaving filter}}{\text{Weight of dust fed to filter}} \times 100 = \text{arrestance per cent.}$$

The accuracy by which the amount of dust in the air leaving the filter could be measured by use of the alundum crucible was checked by running the test apparatus with no filter in the air stream. In these tests the amount of dust as measured by the crucible method showed a maximum variation of approximately 5 per cent from that actually fed into the filter, and the average variation was 1.6 per cent. For a filter of 90 per cent efficiency a variation of 5 per cent in the weight of dust leaving the filter would give a maximum possible error of 0.5 per cent in the over-all arrestance of the filter.

PROCEDURE FOR DUST CONCENTRATION TESTS

Filters *A*, *B*, *C*, and *D* were all used in this series. The dust mixture consisted of 50 per cent by weight of Pocahontas ash screened through a 200 mesh screen, 20 per cent by weight of Illinois fly ash screened through a 200 mesh screen, 20 per cent by weight of lampblack (Germantown Eaglebrand) screened through a 100 mesh screen, and 10 per cent by weight of Fuller's earth, all thoroughly mixed and screened through a 100 mesh screen. A face air velocity of 300 fpm was selected, and the dust mixture was fed into the air stream at the rates of 10, 20, 30, 40, 60, and 100 g of dust per hour. This gave dust concentrations of 0.2, 0.4, 0.6, 0.8, 1.2 and 2.0 g per 1000 cu ft of air respectively. For those filters in which it was evident from preliminary test data that small variations in the dust concentration would have no great effect on the filter arrestance, a part of the specified tests was omitted. In all of the tests the required amount of dust was fed into the air stream leading to the filter at a constant uniform rate, and the arrestances were taken at the

TABLE 3—FILTER *A*. TEST RESULTS FOR DIFFERENT DUST CONCENTRATIONS
(*Face Air Velocity* 300 *fpm*. *Dust mixture* 50, 20, 20, 10)

TEST NO.	RATE OF FEED	INITIAL RESIS.	LENGTH OF TEST	GRAM-HOURS	AVG ARREST-ANCE	DUST-HOLDING CAPACITY
16	20	0.225	32.8	656	77.2	503.8
134	40	0.229	14.5	580	77.5	449.0
111	40	0.220	15.1	604	80.3	483.6
112	60	0.220	11.0	657	76.9	504.8
144	100	0.230	5.2	520	79.7	411.9

end of each one-hour period. On this basis each test was continued until the air pressure drop across the filter reached 0.4 in. of water.

The results of these tests are shown in Tables 3 to 8, inclusive, and in Figs. 7, 8, 9, and 10 for Filters *A*, *B*, *C*, and *D*, respectively. In order that the curves for each filter might be on a comparable basis the air resistances across the filter and percentage arrestance were plotted against weight of dust fed to the filter in grams. In each case the points only are shown for the arrestance curves at feeds of 10 g per hour. Due to the irregularity of these points the curves were omitted. This irregularity was due to the difficulty of determining the small increases in crucible weights necessarily accompanying a low rate of feed. Two crucibles were used alternately throughout all tests.

Referring to the figures in Table 3 and the curves of Fig. 7, which record the performance data for Filter *A*, it will be noted that there are slight variations in arrestance but no general trend to show an advantage for either the low or high rates of dust feed. The highest dust-holding capacities were for the tests in which the rates of feed were 20 and 60 g per hour. The highest arrestance was for Test No. 3 at 40 g per hour. The dust-holding capacity dropped off materially at a rate of feed of 100 g per hour, indicating that the filter was overloaded at this rate of feed. From 20 to 60 g per hour, however,

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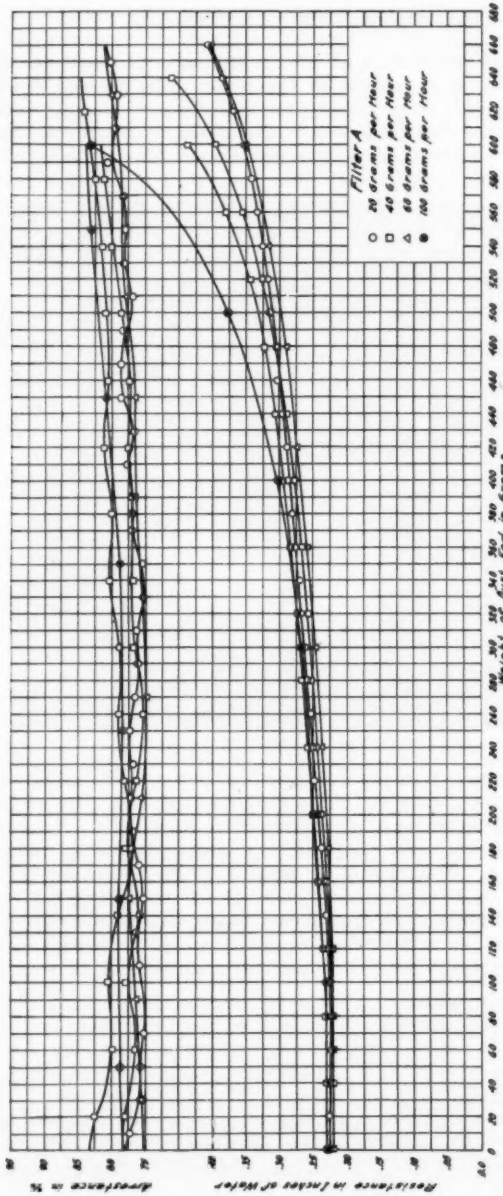
Filter A
20 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour
Weight of Dust Fed in Grams
Resistance in Inches of Water
Arrestance in %

FIG. 7. ARRESTANCE AND RESISTANCE CURVES AT VARIOUS RATES OF DUST FEED FOR FILTER *A*

TABLE 4—FILTER *B*. TEST RESULTS FOR DIFFERENT DUST CONCENTRATIONS
(*Face Air Velocity* 300 *fpm*. *Dust mixture* 50, 20, 20, 10)

TEST NO.	RATE OF FEED	INITIAL RESIS.	LENGTH OF TEST	GRAM-HOURS	AVG ARREST-ANCE	DUST-HOLDING CAPACITY
106	10	0.146	24.8	248	80.0	199.8
12	20	0.153	15.8	316	78.8	249.2
102	30	0.150	9.6	288	81.3	233.6
104	40	0.148	8.0	320	81.0	259.2
103	40	0.152	7.3	292	81.9	237.5
101	40	0.150	8.7	347	81.7	282.4
107	60	0.157	5.4	324	82.0	264.6
143	100	0.147	3.9	390	82.8	322.8

the performance of the filter may be considered equally satisfactory for any rate. The dust-catching surfaces of the filter were well graded throughout the filter and at rates of dust feed below 60 g per hour there did not seem to be a tendency to overload the filter.

The test results for Filter *B* are recorded in the figures of Table 4 and by the curves of Fig. 8. From these results it would appear that the performance of the filter improves as the rate of dust feed is increased. However, there is a variation in the test results for different filters under the same test conditions as noted in Tests Nos. 104, 103, and 101, all of which were run at a dust concentration of 40 g per hour on filters which had substantially the same initial resistance. In general, the performance at rates of feed from 30 to 60 g of dust per hour checked within reasonable limits. The low dust-holding capacity at 10 g and high dust-holding capacity at 100 g would indicate a better distribution of dust throughout the filter at the high rates of feed than at the very low rates of feed.

The test results for Filter *C* are shown in the figures of Table 5 and the curves of Fig. 9. In this filter, as in Filter *B*, the general performance characteristics tend to improve with the higher rates of dust feed. The dust-holding capacity increases directly with the rate of feed, and while there is a slight tendency for a reduction in dust arrestance at the higher rates, this reduction is not great and the average arrestance only drops from 72.3 per cent at

TABLE 5—FILTER *C*. TEST RESULTS FOR DIFFERENT DUST CONCENTRATIONS
(*Face Air Velocity* 300 *fpm*. *Dust Mixture* 50, 20, 20, 10)

TEST NO.	RATE OF FEED	INITIAL RESIS.	LENGTH OF TEST	GRAM-HOURS	AVG ARREST-ANCE	DUST-HOLDING CAPACITY
145	10	0.142	29.0	290	72.3	210.7
11	20	0.148	15.6	312	72.0	221.0
105	40	0.142	9.0	358	69.9	250.3
127	60	0.156	6.0	360	71.4	256.0
142	100	0.147	4.6	460	68.2	313.9

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TABLE 6—FILTER *D*. TEST RESULTS FOR DIFFERENT DUST CONCENTRATIONS
(*Face Air Velocity* 300 *fpm*. *Dust Mixture* 50, 20, 20, 10)

TEST NO.	RATE OF FEED	INITIAL RESIS.	LENGTH OF TEST	GRAM-HOURS	AVG ARREST-ANCE	DUST-HOLDING CAPACITY
110	10	0.086	20.7	207	89.8	186.2
31	20	0.088	10.8	216	92.0	198.5
113	20	0.091	11.4	228	92.5	208.8
108	40	0.095	6.3	252	93.5	233.5
109	60	0.090	4.1	246	94.6	231.1
149	100	0.087	2.4	240	92.4	222.5

TABLE 7—AVERAGE DUST ARRESTANCES FOR FILTERS *A*, *B*, *C*, AND *D*
(*Face Air Velocity* 300 *fpm*. *Dust Mixture* 50, 20, 20, 10)

FILTER	RATE OF DUST FEED—GRAMS PER HOUR					
	10	20	30	40	60	100
(*A*)	...	77.2	...	77.5 80.3	76.9	79.7
(*B*)	80.0	78.8	81.3	81.0 81.9 81.7	82.0	82.8
(*C*)	72.3	72.0	...	69.9	71.4	68.2
(*D*)	89.8	92.5 92.0	...	93.5	94.6	92.4

TABLE 8—AVERAGE DUST-HOLDING CAPACITY UP TO 0.4 IN. PRESSURE DROP FOR FILTERS *A*, *B*, *C*, AND *D*
(*Face Air Velocity* 300 *fpm*. *Dust Mixture* 50, 20, 20, 10)

FILTER	RATE OF DUST FEED—GRAMS PER HOUR					
	10	20	30	40	60	100
(*A*)	...	503.8	...	449.0 483.6	504.8	411.9
(*B*)	199.8	249.2	233.6	259.2 237.5 282.4	264.6	322.8
(*C*)	210.7	221.0	...	250.3	256.0	313.9
(*D*)	186.2	198.5 208.8	...	233.5	231.1	222.5

10 g of dust per hour to 68.2 per cent at 100 g of dust per hour. The dust-holding capacities for the low and the high rates of feed are substantially the same for Filters *B* and *C*.

The test results for Filter *D* are shown in the figures of Table 6 and the curves of Fig. 10. For this filter the dust-holding capacity increased with rate of dust feed from 10 to 40 g per hour with approximately the same capacity at 60 g per hour. At 100 g per hour there is a tendency toward a decrease in dust-holding capacity. Both the arrestance and dust-holding capacity are lower

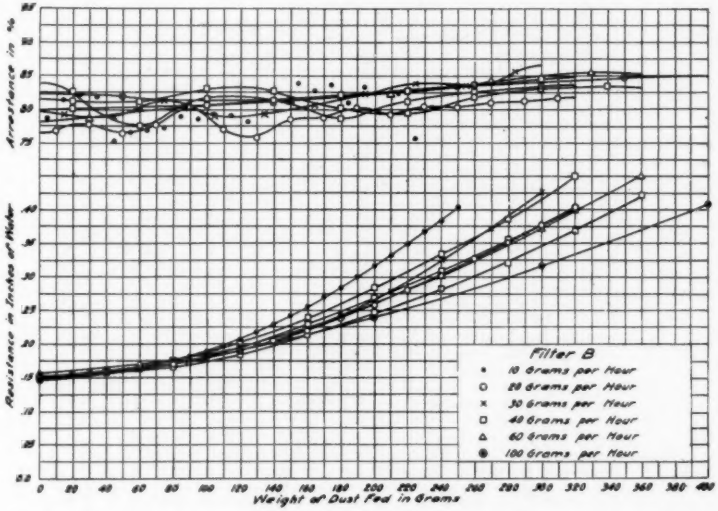
Arrestance in %
Resistance in Inches of Water
Weight of Dust Fed in Grams
Filter B
10 Grams per Hour
20 Grams per Hour
30 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour

FIG. 8. ARRESTANCE AND RESISTANCE CURVES AT VARIOUS RATES OF DUST FEED FOR FILTER *B*

for the 10 g rate than for the higher rates. From 20 to 100 g of dust per hour the arrestances are substantially the same, the highest being at 60 g per hour.

The conclusions which may be drawn from this series of tests are as follows:

1. Within the range of 10 to 100 g of dust feed per hour the rate of feed has no appreciable effect on the arrestance of any of the four types of filters tested. This conclusion is borne out by reference to Table 7 which shows a summary of the average arrestances for all filters at different rates of feed. The spread between the average arrestance values for any given filter at different rates of feed might have been reduced if more tests had been used to determine the average value for each rate of dust feed.

2. As shown in Table 8 there was but very little change in dust-holding capacities in any of the filters due to varying the rate of dust feed from 20 to 60 g per hour. For Filters *D* and *C* there was a marked increase in dust-holding capacities at a rate of 100 g per hour, and for Filters *B*, *C* and *D*, the only ones tested at 10 g of dust per hour, there was a decrease in dust-holding capacity for the 10 g rate. For Filters *B* and *C* the dust-holding capacity at 100 g of dust feed per hour was approximately 50 per cent greater than the dust-holding capacity for 10 g of dust per hour. For Filter *A* the dust-holding capacity at 100 g per hour is a considerable amount

lower than that for any of the feeds between 20 and 60 g per hour. The increase in dust-holding capacity for Filters *B* and *C* at 100 g rate indicated that at this high rate of feed a lesser amount of dust was retained on the entering surface of the filter and therefore more dust was carried back into the filter to give a more uniform

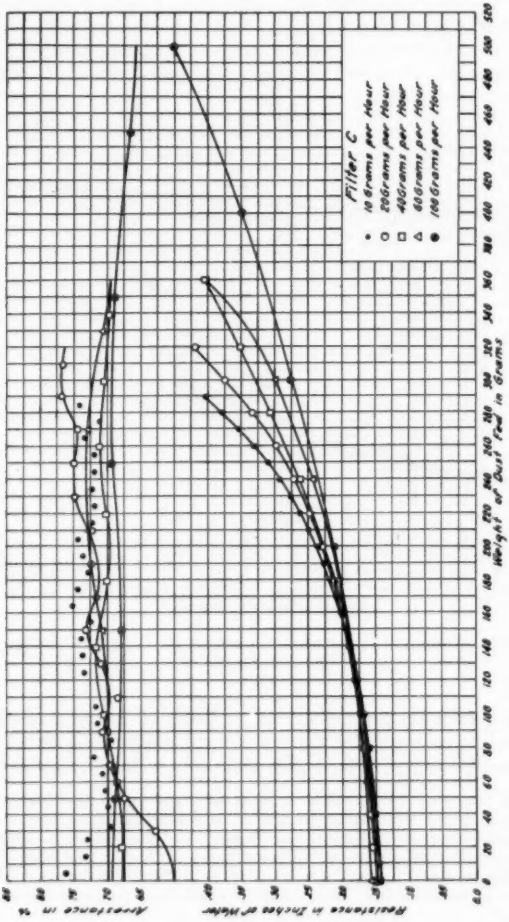
Filter C
10 Grams per Hour
20 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour
Weight of Dust Fed in Grams
Arrestance in %
Resistance in Inches of Water

FIG. 9. ARRESTANCE AND RESISTANCE CURVES AT VARIOUS RATES OF DUST FEED FOR FILTER C

distribution throughout the filter volume, thus making it possible for the filters to retain more dust without building up an excessive air resistance across the filter.

3. Insofar as dust arrestance is concerned, the rate of dust feed for standard air filter rating tests might be set at any value from 10 to 100 g per hour, and insofar as dust-holding capacity is concerned it might be set at any value from 20 to 60 g per hour with substantially the same results for any filters of the types tested.

PROCEDURE FOR AIR VELOCITY TESTS

In this series the four Filters *A*, *B*, *C*, and *D* were tested. The dust mixture used was the same as in the previous series and it was fed into the air stream at the rate of 20 g per hour. Each filter was tested at air velocities of 200, 300, and 400 fpm. The tests were all three hours in length, the arrestance and air resistance being determined for each hour of the test. Since the purpose of the test was to determine the effect of air velocity on the dust

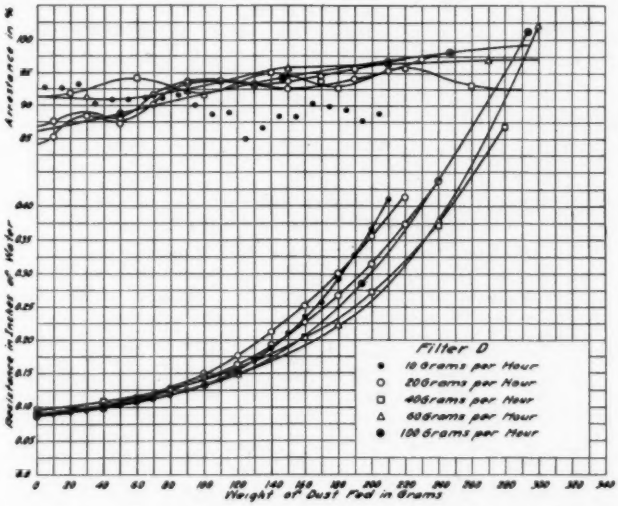
Filter D
10 Grams per Hour
20 Grams per Hour
40 Grams per Hour
60 Grams per Hour
100 Grams per Hour
Weight of Dust Fed in Grams

FIG. 10. ARRESTANCE AND RESISTANCE CURVES AT VARIOUS RATES OF DUST FEED FOR FILTER *D*

arrestance of the filter, it was not considered necessary to run the test for more than a three-hour period.

A summary of the test results is shown in Table 9. These tests show that there is but very little difference in the dust arrestance between the air velocity of 200 and 400 fpm. There is, however, in all cases a slight increase at the 400 fpm air velocity, and in Filter *A* this increase is practically uniform over the range from 200 to 400 fpm air velocity. In Filters *B*, *C*, and *D* the dust arrestance at 300 ft is slightly less than that at 200 fpm air velocity. The increase in arrestance over the range of tests runs from 1.2 per cent to 5.8 per cent for Filters *C* and *D* respectively.

Even though these increases are small they indicate that velocities higher than 300 fpm might be used on any of the filters without impairing their dust-catching arrestance.

FILTER CHARACTERISTICS AS GOVERNED BY CONSTITUENT DUSTS OF MIXTURE

The purpose of this series of tests was to determine the effect of individual dusts used in the dust mixture on the operating characteristics of the filter

TABLE 9—DUST ARRESTANCE AT DIFFERENT AIR VELOCITIES

(*Dust Mixture* 50, 20, 20, 10. *Dust feed* 20 *g per hour*)

AIR VELOCITY IN FPM	FILTER			
	A	*B*	*C*	*D*
200	74.4	77.4	65.7	89.1
300	76.1	77.0	65.0	86.9 87.8
400	77.6	78.7	66.2	92.6
Range (Per Cent Diff.)	3.2	1.7	1.2	5.8

TABLE 10—AVERAGE DUST ARRESTANCE AND INCREASE IN AIR RESISTANCE FOR DIFFERENT TYPES OF FILTERS[a]

(*Dust fed at rate of* 20 *g per hour for a three-hour period*)

TYPE DUST	FILTER *A*		FILTER *B*		FILTER *C*		FILTER *D*	
	Resist. Increase In. of H_2O	Average Arrestance	Resist. Increase In. of H_2O	Average Arrestance	Resist. Increase In. of H_2O	Average Arrestance	Resist. Increase In. of H_2O	Average Arrestance
Illinois Fly Ash	−0.001	94.0	0.011	96.7	0.003	92.1	0.004	84.0
Pocahontas Ash	0.001	92.4	0.009	89.8	0.008	82.5	0.006	86.4
Cottrell Ash	0.001	92.8	0.004	89.9	0.002	82.3	0.006	73.7
Fuller's Earth	−0.004	87.2	0.011	89.8	0.002	81.0	0.006	82.9
Lampblack	0.002	21.7	0.023	27.8	0.022	15.3	0.288	74.1

[a] Average Arrestance and Resistance Increase for three-hour period.

TABLE 11—SPREAD BETWEEN DUST ARRESTANCE VALUES FOR DIFFERENT KINDS OF DUST

(*Air Velocity* 300 *fpm.* 20 *g dust per hour*)

TYPE OF DUST	EFFICIENCY SPREAD		
	High	Low	Per Cent Spread
Illinois Fly Ash	96.7 *B*	84.0 *D*	12.7
Fuller's Earth	89.8 *B*	81.0 *C*	8.8
Pocahontas Ash	92.4 *A*	82.5 *C*	9.9
Lampblack	74.1 *D*	15.3 *C*	58.8
Cottrell Ash	92.8 *A*	73.7 *D*	19.1

and to compare the arrestance values as determined by tests of filters when using various dust mixtures with the weighted values as obtained when using individual dusts of the mixture. In this series of tests all filters were used, the air velocity through the filters was set at 300 fpm, and the dust mixtures were fed to the filters at the rate of 20 g per hour. All tests were four hours in length excepting in those cases where the filter resistance increased to 0.4 in. of water before the end of the four-hour period. The dust arrestance and air resistance were recorded for each one-hour period. The results for these tests are shown in Tables 10, 11, and 12, and in Figs. 11 to 15 inclusive.

The average arrestance values and increase in resistance across the filter for five different types of dust are shown in Table 10. The increase in air re-

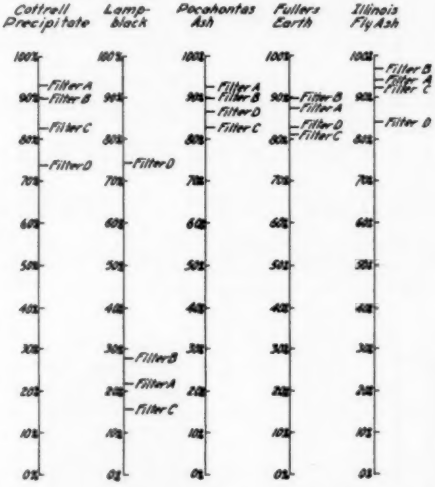

FIG. 11. COMPARATIVE FILTER ARRESTANCES FOR COMPONENT DUSTS

sistance is shown for the three-hour period, and for all types of dust excepting lampblack the increase is small. In the case of Filter *A* there was a slight decrease in resistance when using Illinois fly ash and Fuller's earth. This was probably due to readjustment of oil on the filter media and the streamlining of the oil on the fibers due to passage of air during initial stages of test. The rise in resistance when using lampblack was high in all cases, but this is particularly marked for Filter *D* in which the three-hour increase was recorded as 0.288 in. of water. The dust arrestance for Filter *D* when using lampblack was much higher than for any of the other filters. This partly accounts for the rapid increase in air resistance of Filter *D* as compared with the other filters.

The spread in arrestance values for the different types of dust is shown in the figures of Table 11 and in the chart of Fig. 11. The three greatest spreads were 58.8 per cent for lampblack, 19.1 for Cottrell precipitate, and 12.7 per cent for Illinois fly ash. In each of these cases the wide range is caused

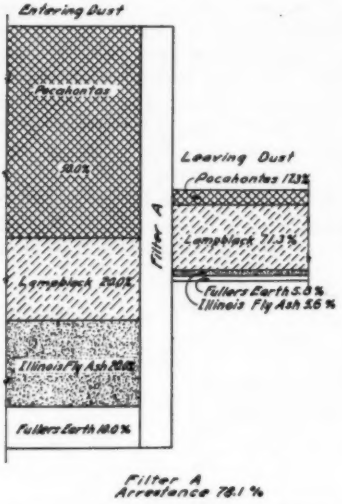

FIG. 12. RELATIVE WEIGHTS OF MIXED DUST ENTERING AND LEAVING FILTER *A*

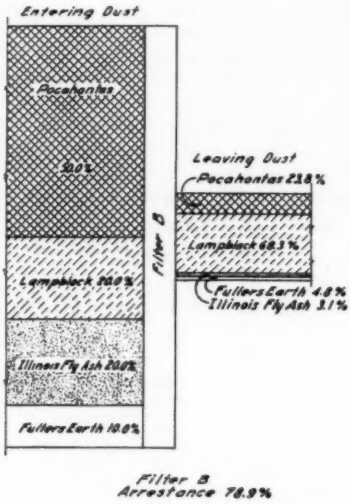

FIG. 13. RELATIVE WEIGHTS OF MIXED DUST ENTERING AND LEAVING FILTER *B*

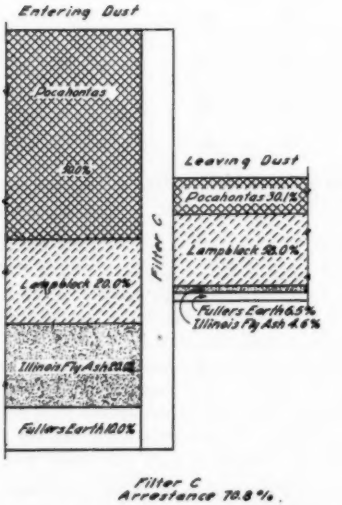

FIG. 14. RELATIVE WEIGHTS OF MIXED DUST ENTERING AND LEAVING FILTER *C*

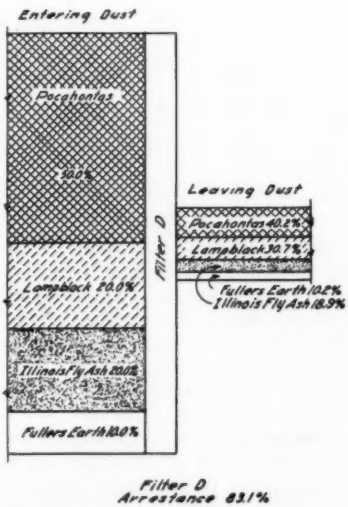

FIG. 15. RELATIVE WEIGHTS OF MIXED DUST ENTERING AND LEAVING FILTER *D*

mostly by the characteristics of Filter *D*. This is clearly shown by the chart of Fig. 11. Filter *D* is not quite as efficient on Cottrell precipitate and Illinois fly ash as the other filters, but is very much more efficient on lampblack. The standard lampblack as used consists of very fine particles which are difficult to remove by the ordinary viscous coated filters. It seems to be removed only by the very fine mesh part of the filter, and when so removed builds up the air resistance in the filter very rapidly. The large percentage (50 per cent) of this material specified in the A.S.H.V.E. Standard Test Code and used in former tests undoubtedly accounts for the low arrestance values obtained for some types of filters.

The arrestance values for all filters were determined for the Standard Code mixture of dust consisting of 50 per cent Pocahontas ash and 50 per cent lampblack. These values were also determined at air velocities of 300 fpm and for 20 g of dust fed to the filter per hour. The arrestance values for these

TABLE 12—COMPARISONS OF ARRESTANCE VALUES AS DETERMINED BY TESTS WITH DUST MIXTURES AND BY WEIGHTED VALUES FROM TESTS ON INDIVIDUAL DUSTS

(*Air Velocity* 300 *fpm.* 20 *g dust per hour*)

TYPE OF FILTER	50% POCAH., 20% ILL. 20% LAMP., 10% FULL.		50% POCAHONTAS 50% LAMPBLACK		PER CENT DIFFERENCE	
	Straight Mix Test	Weighted Values	Straight Mix Test	Weighted Values	50–20–20–10	50–50
(*A*)	76.1	78.1	54.6	57.0	+2.0	+2.4
(*B*)	77.0	78.9	56.2	58.9	+1.9	+2.7
(*C*)	65.0	70.8	46.5	48.9	+5.8	+2.4
(*D*)	87.7	83.1	81.5	80.3	−4.6	−1.2
	86.9				−3.8	

tests and the arrestance values for the 50 per cent, 20 per cent, 20 per cent, 10 per cent dust mixture tests were compared with the weighted arrestance values as calculated from the test values for each individual dust on the same filter. The weighted values were calculated by multiplying the test values for each separate dust by the percentage of that dust in the final mixture and adding the results for all dusts of the mixture. These comparative values are shown in Table 12. It is interesting to note that in general there is a very close agreement between the calculated weighted values and the test values for the different dust mixtures. This means that any dust in the mixture will affect the performance characteristics of the filter in direct proportion to the percentage of that dust in the mixture. Since the lampblack which has been in common use for testing filters according to the Standard Code is a very difficult dust for most filters to remove, and since, when efficiently removed, it builds up the filter resistance very rapidly, it is evident that the large percentage specified in the Standard Test Code severely handicaps the performance of most filters either by lowering their dust arrestance values or by increasing the air resistance values and thus shortening the life of the filter.

Figs. 12, 13, 14, and 15 show in diagram form the performance charac-

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teristics of Filters *A*, *B*, *C*, and *D* when using the dust mixtures indicated. The percentage values shown for the dust before and after passing through the filter indicate the percentages of the individual type of dust in the initial and final weights of dust. These diagrams show very clearly the effect of different types of dust in the mixture on the arrestance of a given filter. For instance, for Filters *A*, *B*, and *C* there is a large percentage of lampblack in the final dust. This explains the low arrestance values for the 50 per cent mixtures as shown in Table 12. In Filter *D* there is a small percentage of lampblack in the final dust, thus explaining the high arrestance for this filter as shown in Table 12. The resistance, however, as shown by Table 10, would be built up very rapidly by an excess of lampblack.

CONCLUSIONS

The final conclusions from this investigation which may be applied to the Test Code are as follows:

1. For rating purposes it would be practical to use heavier dust concentrations than now specified by the Code. From the first series of tests it is apparent that dust concentrations from 0.6 to 1.2 of dust per 1000 cu ft of air would give test results which would be equally as valuable as those at dust concentrations of 0.35 g per 1000 cu ft as specified in the Code. Higher concentrations would greatly expedite any tests to determine the life of a filter. There seems to be some question as to the arrestance, and particularly the life of a filter at very low dust rates. However, any tests to determine the life of a filter at a rate of dust feed comparable to that which may be met in practice would require unlimited time and would be impractical.

2. The air velocity at which a filter is tested does not appear to be important, providing it is maintained between 200 and 400 fpm. Further investigations should be made if other velocities are to be used. There seems to be no practical reason for changing the 300 fpm velocity as specified in the present Code, as this would not particularly expedite the tests.

3. Lampblack is a very difficult dust to remove from the air and, if mixed in the original dust in large quantities, it will unduly handicap the performance of most filters, either by reducing the dust arrestance value or by increasing the air resistance and thus shortening the life of the filter. It is believed that 50 per cent of lampblack is very much in excess of that found in practice and it would seem reasonable to reduce this percentage in the Standard Test Code.

4. The performance of a filter varies greatly with the mixture of dust used. By proper choice of a test dust, one type of filter may be shown to have better performance characteristics than the majority of the other filters, whereas another dust mixture may reverse the relative merits. Three of the four filters tested ranked first in arrestance on one particular dust. This does not infer the fourth filter to be inferior in dust arrestance, as further tests on this same type of filter but with a different design, brought its arrestance up among the highest on one particular dust mixture.

These tests were made for the purpose of studying the characteristics of different types of filters when rated by a laboratory test method and to determine whether or not some improvements might be made in the Standard A.S.H.V.E. Test Code requirement. When interpreting the test results it should be remembered that only one design of each type of filter was selected, and that the filters were not all of the same thickness, nor did they represent the only thickness manufactured for the different types. Other designs of some of the types might have been selected which would have changed the relative merits of certain properties of the filters as shown by the test results.

To compare the merits of the different types of filters would require a much more extended series of tests and any attempt to make comparisons from the data presented here would be misleading due to the limited selection from each type. The selections were, however, sufficiently broad to substantiate the recommendations made in regard to changes in test procedure and Code.

There is a need for further investigation along the following lines:

1. The performance of different types of filters should be investigated when using different types of dusts and dusts of different particle size.
2. A further analysis should be made covering the different types of carbon which fulfill the Standard Code requirements.
3. A study should be made to determine the effect of lint on the performance of various types of filters.
4. A more complete analysis should be made of the dust which must be handled in different types of filter installations. From this analysis it may be found necessary to build up more than one type of synthetic dust for adequately rating different types of filters for different classes of installations.
5. A practical field test should be devised which could be used in the field and which could be used parallel with the laboratory rating test.

DISCUSSION

H. C. MURPHY (WRITTEN): The Committee on Air Cleaning and Atmospheric Impurities feels that this paper indicates very clearly the value of the cooperative research at the University of Minnesota. Aside from the valuable data accumulated in these studies the development of an improved dust feeder is of outstanding importance. This new dust feeder has been adopted by many users of the Code procedure and is being considered for incorporation in the pending revision of the test code.

The data developed as to the efficiency of filters against various types of dust will be of considerable value to the Committee in considering the proposed changes in the make-up of the standard test dust.

These tests indicate again that lampblack is one of the most difficult substances to remove from an air stream. Unquestionably the proportion of lampblack carbon in the standard test dust is, as pointed out by these investigations, considerably higher than that found in ordinary city air. There is consequently no doubt that efficiencies arrived at under the standard A.S.H.V.E. Code are definitely lower than can be expected from the same cleaning device when installed and operated in an actual ventilating or air conditioning system.

It must be emphasized, however, that the present A.S.H.V.E. Code, as stated in its preamble, is "for the laboratory investigation and rating" of air cleaning devices. The Code further states, "Ratings established under this Code are not to be confused with operating efficiencies in actual service—the dust concentration, the nature of the dust, the relative humidity and many other factors have a definite bearing upon the results obtained in actual service".

As originally conceived, this particular Code was to be merely a yardstick for the laboratory comparison of various air cleaning devices. The Committee has been endeavoring to develop a practical test code for use in field investigations, and a favorable outcome now seems possible.

While it may prove advisable to reduce somewhat the lampblack content in the standard test dust, the value of this Code, it is believed, would be lessened if the test dust were changed to one which is easily handled by all air cleaners. If the test dust, for instance, were made of particles the size of peas, all air filters, whether good or bad air cleaners, would show an arrestance of 100 per cent and it would be impossible to grade them one against the other. Professor Rowley's test shown in

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Fig. 11, in which the filters were tested against straight Illinois fly-ash, brings out this point very clearly.

Using this test dust, all the filters had arrestances between 85 per cent and 98 per cent, whereas using straight lampblack these same filters had arrestances varying from as low as 15 per cent to as high as 75 per cent.

It is entirely probable, as Rowley and Jordan have pointed out in this paper, that it will be desirable to rate filters against more than one type of dust. It is obvious, for instance, that the filters on the recirculation intake of a department store will have an entirely different problem than the filters on the fresh air intake. The recirculated air will probably have a much higher lint content than the fresh air, and it may be advisable for the engineer to select the type of cleaner to be used with these considerations in mind.

R. S. DILL[1]: I noted during the presentation of this paper that the effectiveness of filters was evaluated in percentages efficiency, carried to three significant figures. Rather than comment on the paper, I question whether the third digit is justified by the precision of the test method and if so, what is the limit of precision of the apparatus and procedure?

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[1] Assoc. Mech. Engr., National Bureau of Standards, Washington, D. C.

No. 1095

STUDY OF SUMMER COOLING IN THE RESEARCH RESIDENCE USING A SMALL CAPACITY MECHANICAL CONDENSING UNIT

By A. P. KRATZ,* S. KONZO,** M. K. FAHNESTOCK,*** AND E. L. BRODERICK †
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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois.

THE previous investigations in summer cooling in the Research Residence during the summers of 1932 to 1936 inclusive made use of central cooling plants of approximately 30,000 Btu per hour capacity, employing either ice, mechanical refrigeration, or cold water. The investigation for the summer of 1937 was undertaken to determine the temperature and humidity conditions that could be maintained in the Research Residence with a mechanical condensing unit having a capacity approximately 40 per cent less than the 30,000 Btu per hour previously used; particularly when such cooling was supplemented by the circulation of outdoor air through the house at night, and when approximately one air change per hour of outdoor air was used for the purpose of ventilation during the day.

In the first series of tests (Series 1-37) made during the summer of 1937, cooled air was distributed to the rooms on both the first and second stories when the indoor air conditions were such that cooling was required. However, in the second series of tests (Series 2-37) the cooled air was delivered only to the rooms on the first story during the daytime, and only to the rooms on the second story after 6:30 p. m.

DESCRIPTION OF THE RESEARCH RESIDENCE AND COOLING EQUIPMENT

The Research Residence together with the forced-air heating system has been described in a previous paper.[1] For the purpose of this investigation the Resi-

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† Research Assistant, Engineering Experiment Station, University of Illinois.

[1] Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 95).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by M. K. Fahnestock.

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dence was equipped with awnings at all east, south, and west windows, and the sunroom was isolated from the rest of the house by closing the doors leading into the dining room. The entire third story was regarded as an attic and

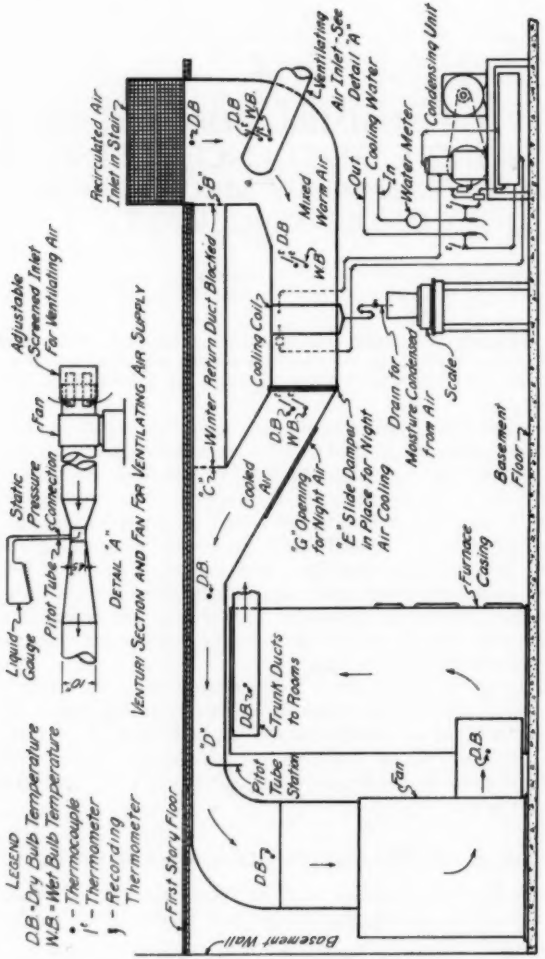


FIG. 1. DIAGRAM OF COOLING PLANT WITH MECHANICAL REFRIGERATION

during the daytime was isolated from the rest of the house by closing the door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic both day and night. The total space cooled amounted to

14,170 cu ft. Unless otherwise specified, the state of the Residence was the same as that during previous summers.

With the exception that the duct leading to the kitchen register was connected to the south trunk duct which supplied air to the first story rooms only, the arrangement of the forced-air duct system and fan was practically the

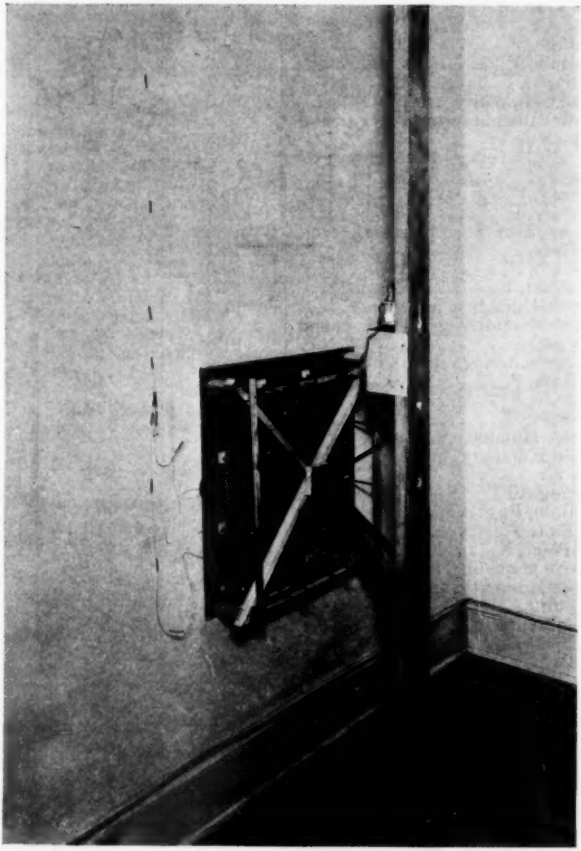


FIG. 2. HEAT-FLOW METER INSTALLATION ON NORTH WALL

same as that used in previous summers. For the first series of tests (Series 1-37), in which both the first and second story rooms were cooled at the same time, the dampers in the ducts to the first and second stories were adjusted to maintain the proper balance between the amounts of air required for cooling on these two stories. For the second series of tests (Series 2-37), in which

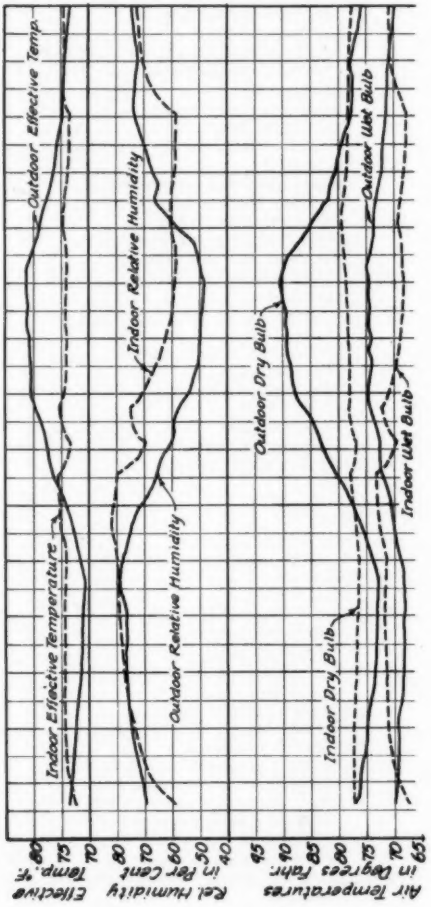
TABLE 1—TYPICAL OPERATING DATA AND RESULTS OBTAINED WITH

1. Date	9–1–37	8–17–37	8–17–37
2. Test No.	25	19	19
3. Series	1–37	2–37	2–37
4. Time of Day	2:00 P.M.	3:30 P.M.	7:30 P.M.
5. Outdoor Air			
Dry-Bulb, F	89.3	92.8	80.3
Wet-Bulb, F	74.3	75.8	71.3
Dew-Point, F	68.2	69.1	67.5
Relative Humidity, per cent	50.0	46.0	65.0
Specific Humidity, grains per pound dry air	103.8	107.0	101.2
6. Indoor Air, average breathing level temp.			
1st Story, F	78.4	78.1	79.4
2nd Story, F	78.1	83.0	81.0
7. Indoor Air, average at breathing level	Both Stories	1st St.	2nd St.
Dry-Bulb, F	78.2	78.1	81.0
Wet-Bulb, F	69.0	66.7	68.8
Dew-Point, F	64.6	60.7	63.0
Relative Humidity, per cent	62.9	55.0	54.0
Specific Humidity, grains per pound dry air	91.6	79.6	86.0
8. Indoor Air, entering return grille			
Dry-Bulb, F	78.0	76.6	79.2
Wet-Bulb, F	69.0	68.1	68.8
Dew-Point, F	64.8	63.9	63.8
Relative Humidity, per cent	64.0	64.8	59.0
Specific Humidity, grains per pound dry air	92.1	89.2	88.8
9. Ventilating Air			
Dry-Bulb, F	88.1	90.1	79.9
Wet-Bulb, F	75.1	74.1	72.7
Dew-Point, F	69.9	67.4	69.7
Relative Humidity, per cent	55.0	47.5	71.0
Specific Humidity, grains per pound dry air	110.4	101.0	109.6
10. Mixed Air, entering cooling coil			
Dry-Bulb, F	79.8	79.5	78.7
Wet-Bulb, F	70.2	69.7	69.4
Dew-Point, F	66.0	65.2	65.1
Relative Humidity, per cent	62.5	61.5	63.0
Specific Humidity, grains per pound dry air	95.8	93.4	93.2
11. Mixed Air, leaving cooling coil			
Dry-Bulb, F	68.7	67.2	67.3
Wet-Bulb, F	64.5	62.8	63.4
Dew-Point, F	62.3	60.3	61.2
Relative Humidity, per cent	82.0	78.5	81.0
Specific Humidity, grains per pound dry air	84.2	78.4	81.2
12. Air Temp. drop through cooling coil, F	11.1	12.3	11.4
13. Temp. cooled air leaving registers, average, F	72.0	70.5	72.0
14. Air Temp. rise in ducts and casing F	3.3	3.3	4.7
15. Basement Air Temp. at breathing level, F	74.2	74.8	74.2

SMALL-SIZED CONDENSING UNIT ON AUGUST 17 AND SEPTEMBER 1, 1937

	Both Stories	1st St.	2nd St.
16. Quantity of air through wet coil			
Cubic feet per minute	978	916	958
Pounds per hour	4,400	4,120	4,310
Density or air, pounds per cubic foot.	0.075	0.075	0.075
17. Number of air recirculations per hour	4.1	7.8	8.0
18. Ventilating Air			
Cubic feet per minute	231	228	229
Pounds per hour	1,040	1,026	1,032
Density of air, pounds per cubic foot.	0.075	0.075	0.075
19. Cooling Coil			
Gross Face Area, square feet	2.07	2.07	2.07
Net Free Area, square feet	1.02	1.02	1.02
20. Air Velocity through Coil			
Gross Face, feet per minute	473	443	463
Free Area, feet per minute	959	898	940
21. Moisture condensed from air, pounds per hour	6.16	5.47	5.95
22. Heat given up by air; total Btu per hour	18,190	17,900	18,050
Heat due to moisture in air			
Btu per hour	6,470	5,740	6,250
Per cent of total heat	35.5	32.0	34.6
Sensible heat			
Btu per hour	11,720	12,160	11,800
Per cent of total heat	64.5	68.0	65.4
23. Water Temp. in condenser			
Inlet, F	59.7	59.8	59.5
Outlet, F	102.7	101.8	101.4
Rise, F	43.0	42.0	41.9
24. Water Temp. in house			
Inlet, F	59.5	59.6	59.5
Outlet, F	102.7	101.5	101.0
Rise, F	43.2	41.9	41.5
25. Quantity of cooling water			
Pounds per hour	448	442	451
Gallons per hour	53.7	53.1	54.1
26. Heat absorbed by cooling water, Btu per hour			
Through condenser	19,260	18,560	18,900
Through house	19,350	18,520	18,710
27. Refrigerant pressures			
Suction pounds per square inch	40	39	40
Discharge pounds per square inch	131	130	129
28. Nominal Rating of condensing unit with 55 F return refrigerant, 59 F condensing water and 74 F ambient air, Btu per hour	16,300	16,300	16,300
29. Compressor Speed, revolutions per minute	255	255	255
30. Compressor Motor Data			
Size, horsepower	2	2	2
Measured power rate, Kilowatt	1.37	1.34	1.34
31. Fan Speed, revolutions per minute	451	451	451
32. Fan Motor Data			
Size, horsepower	¾	¾	¾
Measured power rate, Kilowatt	435	440	440

the first story rooms were cooled until 6:30 p. m., and the second story rooms were cooled after 6:30 p. m., a damper arrangement was installed in the bonnet of the furnace casing so that the circulating air could be distributed either



entirely through the registers on the first story or entirely through the registers on the second story.

The arrangement of the cooling plant is shown in Fig. 1. The condensing unit was self contained and consisted of a water-cooled shell and finned coil condenser-receiver, and a two-cylinder motor-driven compressor. Cooling was accomplished by direct expansion of dichlorodifluoromethane in an evap-

orator placed in a by-pass in the central cold air return duct. The evaporator, or cooling coil, consisted of finned copper tubing placed two rows deep in the direction of air flow and nine rows high measured along the vertical axis of

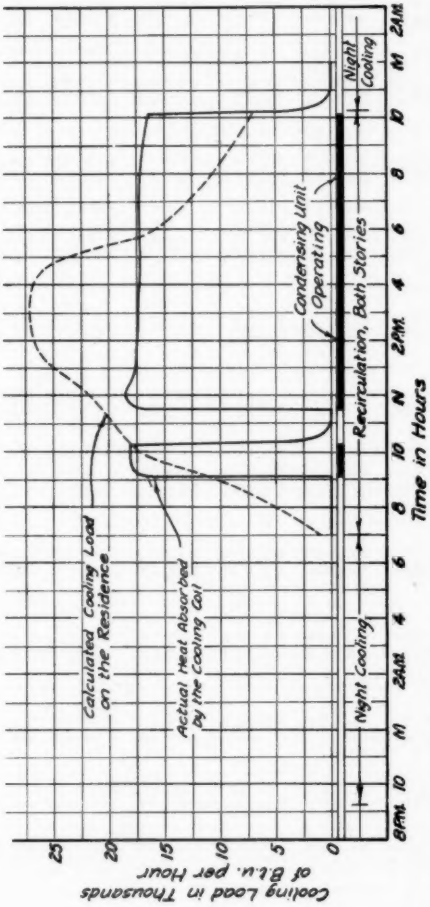


FIG. 3. ACTUAL AND CALCULATED COOLING LOAD ON RESIDENCE AND AIR TEMPERATURES. TEST NO. 25, SERIES 1-37, SEPTEMBER 1, 1937

the air duct. The tubes were ¾ in. outside diameter and had fins, eight to the inch, which were continuous strips of sheet metal extending from the top row to the bottom row of tubes. The evaporator as installed in the duct was 14⅝ in. high and 20⅜ in. wide, and the gross, or face area presented to the flow of air was 2.07 sq ft, while the net, or free area was 1.02 sq ft. The nominal capacity of the condenser and evaporator units under the conditions

XUM

TABLE 2—COMPLETE DATA AND RESULTS FOR TESTS

DATE 1937	TEST No.	SERIES No.	WEATHER DATA							INDOOR AIR CONDITIONS				TEMPERATURE DIFFERENCE	
			Degree-Hours		Total Sunshine in Hours and Minutes	Outdoor Air				At End of Night Cooling		Average During Period Requiring Cooling		Outdoor Max. to Average Indoor F	Average Outdoor to Avg. Indoor F
			Above 85 F	Above 90 F		Maximum Temp. During Test F	Minimum Temp. Previous to Test F	Average Temp. During Period Requiring Cooling							
								Temp. F	Hum. Per Cent	Temp. F	Hum. Per Cent	Temp. F	Hum. Per Cent		
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
7–6	1	1a–37	19.4	0	15:00	89.0	65.0	84.3	47.9	73.7	71.0	79.5	54.0	9.5	4.8
7–7	2	1a–37	34.9	0.3	14:49	90.0	67.0	87.0	48.1	75.3	68.5	79.3	51.6	10.7	7.7
7–8	3	1a–37	19.0	0.1	7:15	90.0	75.0	86.4	50.2	76.4	69.6	78.7	58.9	11.3	7.7
7–9	4	1a–37	19.6	0	9:47	90.0	69.0	85.2	58.0	75.4	78.5	79.0	59.6	11.0	6.2
7–10	5	1a–37	45.1	4.0	13:07	91.0	70.0	87.9	56.8	77.1	79.4	79.4	58.0	11.6	8.5
7–11	6	1a–37	11.5	0	13:07	86.0	71.0	84.4	63.2	77.8	75.5	79.2	59.3	6.8	5.2
7–12	7	1a–37	2.1	0	5:32	84.0	70.0	79.0	72.0	77.8	74.3	78.9	66.3	5.1	0.1
7–14	8	1–37	23.6	0	14:28	88.0	68.0	86.0	66.9	76.4	87.3	79.6	65.0	8.4	6.4
7–15	9	1–37	22.6	0	10:55	88.0	73.0	86.1	62.6	78.3	78.8	79.8	61.2	8.2	2.7
7–16	10	1–37	25.6	0	11:18	89.0	72.0	86.2	58.4	78.0	78.1	79.7	59.9	9.3	6.5
7–24	11	1–37	16.6	0	7:58	88.0	67.0	85.9	57.8	75.0	70.5	78.6	59.7	9.4	7.3
8–4	12	1–37	0.0	0	1:35	83.0	68.0	80.7	72.9	75.4	74.6	79.0	72.1	4.0	1.7
8–6	13	1–37	1.0	0	11:48	85.0	68.0	82.5	58.8	73.8	68.3	78.5	60.0	6.5	4.0
8–7	14	1–37	33.9	0.7	14:07	91.0	65.0	87.5	56.4	73.8	74.1	79.1	65.3	11.9	8.4
8–9	15	1–37	0.6	0	6:46	88.0	68.0	83.3	60.7	76.1	81.8	78.8	61.4	9.2	4.5
8–10	16	1–37	13.3	0	10:10	90.0	66.0	84.3	60.9	74.4	75.5	79.1	60.6	10.9	5.2
8–11	17	1–37	35.6	0.6	9:40	91.0	70.0	87.8	58.0	76.5	77.0	79.7	62.0	11.3	8.1
8–16	18	1–37	35.5	1.2	13:47	94.0	65.0	85.9	55.6	73.1	71.9	79.2	60.5	14.8	6.7
8–17	19	2–37	47.1	12.8	11:34	95.0	72.0	86.7	55.4	77.2	77.9	78.6	59.6	16.4	8.1
8–18	20	2–37	19.4	0	8:42	91.0	68.0	86.7	59.4	75.3	63.3	78.5	66.6	12.5	8.2
8–19	21	2–37	27.0	0	13:40	92.0	71.0	86.5	50.6	76.1	73.5	78.8	59.2	13.2	7.7
8–20	22	2–37	21.0	0	12:41	90.0	67.0	84.7	59.3	75.4	68.5	78.7	61.2	11.3	6.0
8–30	23	2–37	12.6	0	9:51	91.0	68.0	83.2	59.4	75.6	73.0	78.5	61.0	12.5	4.7
8–31	24	1–37	6.7	0	6:15	87.0	69.0	83.7	60.4	76.2	76.1	77.7	63.3	9.3	6.0
9–1	25	1–36	27.1	0.3	8:45	92.0	72.0	85.2	58.6	77.0	81.6	78.1	66.5	13.9	7.1
9–2	26	1–37	23.6	0	8:35	91.0	72.0	85.7	53.9	77.2	78.8	78.8	60.5	12.2	6.9
9–3	27	1–37	12.6	0	9:57	89.0	70.0	84.2	58.6	76.4	80.2	78.1	62.0	10.9	6.1
9–4	28	1–37	0.5	0	5:15	87.0	72.0	81.3	65.6	77.0	78.3	77.6	65.2	9.4	3.7

TABLE 2 (*Continued*)—COMPLETE DATA AND RESULTS FOR TESTS

OPERATION OF THE FAN						OVERALL TEST PERIOD			COMPRESSOR OPERATION					TOTAL MOISTURE CONDENSED LB	HEAT ABSORBED BY COOLING COIL DURING OVERALL TEST PERIOD BTU			RATIO MOISTURE TO TOTAL LOAD
Night Air Cooling Previous to Test		Recirculation Previous to Test		Recirculation During Test Period		Start of Cooling with Condensing Unit to Start of Night Air Cooling			Time in Hours	Power Input Kwhr	Condenser Water							
											Quantity in Gallons	Temperature						
Duration Hours	Power Kwhr	Duration Hours	Power Kwhr	Duration Hours	Power Kwhr	Start	End	Duration Hours				Inlet F	Rise F		Load Due to Moisture	Sensible	Total	
17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35
11.50	5.7	9.95	4.2	3.05	1.5	4:57 P.M.	8:00 P.M.	3.05	3.05	4.6	193	58.9	42.9	14.02	14,720	42,350	57,070	0.258
11.00	4.9	8.78	2.8	6.72	3.1	1:47 P.M.	8:30 P.M.	6.72	6.72	10.3	418	58.8	42.5	34.14	35,850	93,900	129,750	0.276
10.50	4.8	5.00	2.1	4.25	2.0	12:00 M	4:15 P.M.	4.25	3.52	5.7	223	58.9	42.3	24.66	25,890	46,750	72,640	0.356
14.75	6.8	5.50	2.2	7.50	3.5	12:30 P.M.	8:00 P.M.	7.50	6.28	10.1	416	59.0	42.1	47.31	49,680	76,150	125,830	0.395
11.00	5.0	3.28	1.3	12.22	5.4	10:17 A.M.	10:30 P.M.	12.22	12.22	19.1	816	58.9	41.8	92.18	96,790	153,500	250,290	0.387
8.50	3.8	3.68	1.7	11.32	5.0	10:41 A.M.	10:00 P.M.	11.32	11.32	17.9	757	58.9	41.3	84.70	88,940	140,900	229,840	0.387
9.00	4.1	2.50	1.0	7.50	3.3	9:30 A.M.	5:00 P.M.	7.50	3.68	5.8	246	59.1	41.7	29.84	31,330	44,000	75,330	0.416
10.10	4.4	5.07	2.3	10.20	4.4	12:04 P.M.	10:16 P.M.	10.20	10.20	13.8	600	59.2	41.4	80.66	84,690	100,300	184,990	0.458
8.73	3.9	1.73	0.9	12.27	5.3	8:44 A.M.	9:00 P.M.	12.27	12.27	16.3	713	59.4	41.0	87.61	91,990	126,800	218,790	0.421
10.00	4.6	2.63	1.1	12.12	5.4	9:38 A.M.	9:45 P.M.	12.12	12.12	16.1	684	59.1	41.4	73.85	77,540	133,100	210,640	0.368
10.67	4.8	6.87	2.9	5.00	2.3	2:17 P.M.	7:17 P.M.	5.00	5.00	6.7	268	59.3	42.0	27.23	28,600	52,550	81,150	0.353
11.83	5.2	10.93	4.9	1.27	0.7	5:56 P.M.	7:12 P.M.	1.27	1.27	1.8	603	59.5	42.0	9.52	10,000	14,900	24,900	0.402
13.25	5.9	9.77	4.3	2.65	1.1	4:46 P.M.	7:25 P.M.	2.65	2.65	3.5	139	59.3	43.1	14.07	14,770	31,150	45,920	0.322
11.58	5.2	6.87	3.0	6.22	2.9	1:52 P.M.	8:05 P.M.	6.22	4.85	6.6	262	59.5	42.4	29.21	30,680	53,200	83,880	0.366
12.42	5.5	7.18	3.3	5.32	2.2	2:11 P.M.	7:30 P.M.	5.32	5.32	7.2	299	59.5	41.1	31.04	32,620	61,100	93,720	0.348
11.50	5.5	8.25	3.4	5.50	2.3	3:15 P.M.	8:45 P.M.	5.50	5.25	7.0	286	59.6	42.2	30.07	31,580	62,500	94,080	0.336
10.25	4.6	5.00	2.2	8.25	3.6	12:00 M	8:15 P.M.	8.25	8.25	11.3	459	59.4	42.7	52.44	55,100	91,300	146,400	0.377
12.17	5.5	8.50	3.7	4.50	2.0	3:30 P.M.	8:00 P.M.	4.50	4.50	5.9	244	59.5	42.3	25.72	27,000	52,900	79,900	0.338
11.00	5.1	3.08	1.2	11.92	5.3	10:05 A.M.	10:00 P.M.	11.92	10.53	14.1	569	59.6	41.9	58.89	61,820	122,100	183,920	0.336
9.00	4.2	5.50	2.3	5.00	2.1	12:30 P.M.	5:30 P.M.	5.00	2.68	3.6	151	59.4	42.4	15.34	16,110	30,300	46,410	0.347
13.50	6.0	5.25	2.3	8.00	3.6	12:15 P.M.	8:15 P.M.	8.00	5.57	7.5	297	59.5	42.2	25.06	26,300	69,600	95,900	0.274
10.75	4.9	5.50	2.3	8.58	3.8	12:30 P.M.	9:05 P.M.	8.58	5.78	7.8	318	59.9	42.0	32.81	34,480	63,650	98,130	0.352
11.50	5.7	7.15	3.0	7.10	3.1	2:09 P.M.	9:15 P.M.	7.10	5.17	7.0	275	59.8	42.6	27.89	29,290	60,100	89,390	0.328
9.75	4.4	3.00	1.3	11.25	4.9	10:00 A.M.	9:15 P.M.	11.25	11.00	15.0	586	59.4	43.2	69.29	72,750	123,000	195,750	0.372
9.75	4.5	2.12	0.8	13.13	5.7	9:07 A.M.	10:15 P.M.	13.13	11.75	15.7	633	59.4	43.6	72.75	76,390	130,300	206,690	0.370
8.75	4.5	2.17	0.9	11.25	4.9	9:10 A.M.	8:25 P.M.	11.25	11.08	14.9	592	59.4	43.1	62.45	65,600	128,400	194,000	0.338
10.58	4.9	3.00	1.2	12.00	5.2	10:00 A.M.	10:00 P.M.	12.00	11.00	14.6	590	59.3	43.2	66.29	69,600	124,000	193,600	0.360
9.00	4.0	2.50	1.1	6.50	2.8	9:30 A.M.	4:00 P.M.	6.50	6.00	8.0	324	58.9	43.3	40.90	42,950	61,400	104,350	0.412

of actual operation at a compressor speed of 255 rpm, with return refrigerant temperature of 55 F, inlet water temperature of 59 F, ambient air temperature of 74 F, and with an air velocity of approximately 479 fpm across the 2.07 sq ft of face area, was 16,300 Btu per hour.

The controls employed consisted of a thermostatic expansion valve having the thermostatic bulb clamped to the suction line near the junction with the evaporator; a pressure operated water control valve, used to regulate the flow of water through the condenser; and a room thermostat, located in the hall on the second story. The latter served to start and stop the compressor in accordance with the cooling load required to maintain constant room temperature.

The cooling coil was installed in a by-pass in the central return duct, as shown in Fig. 1. For the summer work, the duct was blocked at B and C, and all of the air delivered to the fan in the forced-air system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, E, was placed in the by-pass on the outlet side of the coil, and a door, G, was placed in the recirculating duct on the down-stream side of the slide damper. When outdoor air was required, the basement door and the door, G, in the recirculating duct were opened, and the slide damper, E, was closed. The fan in the forced-air system delivered approximately 1000 cfm of air when recirculating the air in the house and 2220 cfm when using outdoor air at night. These air volumes were based on an air density of 0.075 lb per cubic foot. Outdoor air, for the purpose of ventilation during the day, was provided by means of the duct shown as Detail A in Fig. 1.

The study of the heat flow through exposed west and north walls was made in the northwest bedroom of the Research Residence. Two Nicholls' heat-flow meters [2] were installed, one on the west wall and one on the north wall, with the centers at an approximate height of 33 in. above the floor. The wall section consists of lap siding, building paper, sheathing, studding space, wood lath and plaster with rough sand finish. The coefficient of heat transmission for this uninsulated wall section is 0.25 Btu per square foot per hour per degree Fahrenheit, at a wind velocity of 15 mph. A thin felt pad was placed over the plaster wall and the heat-flow meter was fitted tightly against the felt pad by means of a wooden frame, as shown in Fig. 2. A complete set of 17 thermocouples, in addition to those incorporated in the heat-flow meters, was installed so that air temperatures and surface temperatures both underneath and adjacent to the heat-flow meters could be obtained.

METHOD OF CONDUCTING TESTS

Continuous records were made during the summer of air temperatures in the various parts of the house and in the different sections of the duct system. A complete description of the measurements which were made has been given in a previous report.[3]

During the periods of operation, observations were made of the weight of water circulated through the condenser, the temperatures of the water entering

[2] Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, pp. 65-104).

[3] Study of Summer Cooling in the Research Residence Using Water from the City Water Mains, by A. P. Kratz, M. K. Fahnestock, S. Konzo, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 193).

and leaving both the condensing unit and the house, the head and suction pressures, and the electrical inputs to the compressor and fan motors.

During all of the tests the windows in the attic remained open and those on the first story remained closed. For the purpose of cooling with outdoor air at night, the attic door and 11 windows on the second story were opened in the evening when the effective temperature outdoors became equal to the effective temperature indoors. The dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system until 7 a. m. The fan delivery was 2220 cfm or 9.4 air changes per hour. The windows and doors were closed at 7 a. m. and the dampers were changed so that the fan started delivering recirculated air and the outdoor air admitted

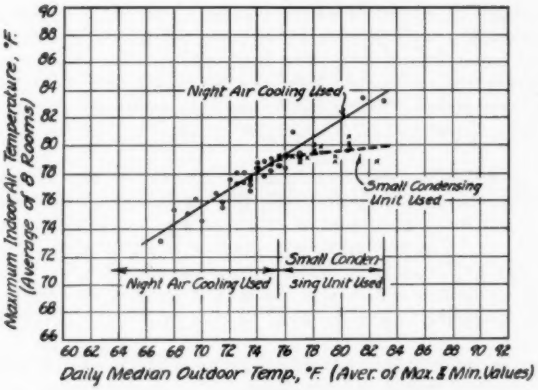


FIG. 4. RELATIONSHIP BETWEEN INDOOR TEMPERATURE AND DAILY MEDIAN OUTDOOR TEMPERATURE

for ventilation. The former was equivalent to 3.2 recirculations of the air in the house and the latter was equivalent to one air change per hour, making a total of 4.2 air changes per hour delivered by the fan. The fan was allowed to run continuously during the day and night.

In the first series of tests, which has been designated as Series 1-37, when the effective temperature of the indoor air on the second story rose to about 75 deg, giving an average of approximately 74.5 deg for both stories, the condensing unit was started, and cooled air was distributed to the rooms on both stories. At the start of the test, when the relative humidity indoors was from 60 to 75 per cent, the corresponding dry-bulb temperature was from 79 to 78 F. The cooling plant was allowed to operate intermittently with the thermostat set to maintain approximately 78 F. However, the heat absorbing capacity of the unit was so small that when the rooms on both stories were being cooled the plant operated practically continuously until the time when cooling with outdoor air at night was started.

In the second series of tests, designated as Series 2-37, the recirculated air was distributed only to the first story rooms after 7 a. m., and the condensing unit was operated intermittently, maintaining an effective temperature of about

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74 deg until 6:30 p. m. At that time the dampers were set so that all of the cooled air was distributed only to the second story rooms. Cooling with outdoor air at night was started when the effective temperature outdoors became equal to the effective temperature of the second story rooms.

Temperatures and heat flow measurements at the heat-flow meter installation were made at regular intervals, both day and night, for a period of 29 days. The data were plotted on a continuous chart and this chart was used as a basis for the analysis of results.

RESULTS OF TESTS WITH SIMULTANEOUS COOLING ON BOTH STORIES

General Conditions

The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on a typical day. For this purpose the test made on September 1, 1937, was selected as typical of the operation obtained when the rooms on both stories were cooled (Series 1-37), and the results are shown in the first column of Table 1 and in Fig. 3. General results from all the tests are given in Table 2.

On the day which was selected as being typical for the tests made under conditions of operation for Series 1-37, the outdoor temperature was 89.3 F at 2 p. m. and reached a maximum of 91.0 F at 4 p. m. In this connection it is of interest to note that the maximum outdoor dry-bulb temperature for the entire season of 1937 was 95 F. The season was considerably cooler than the previous summer season, which was the hottest summer occurring in Urbana, Ill., since the year 1888, when the local Station of the U. S. Weather Bureau was established.

The air conditions at different locations are given in Items 5 to 15 of Table 1. The temperature of the cooled air leaving the registers was approximately 72.0 F. The rise in temperature of the air passing through the furnace casing and ducts was 3.3 F.

The cooling studies made during previous summers [4] had indicated that complete temperature traverses in the duct on both sides of the coil were required in order to obtain the proper corrections to be applied to the averages of the temperatures obtained from the wet- and dry-bulb thermometers at the two reading stations on each side of the coil. The distribution between the sensible and latent heat loads, as shown in Item 22, Table 1, was obtained by using the corrected values for the dry-bulb temperatures, and by multiplying the weighed amount of condensation from the coil by a constant representing the heat given up by the change in moisture content of the air per pound of water vapor condensed. The value of the constant amounted to 1050 Btu per pound of vapor condensed.

Operating Characteristics

The operating characteristics with cooled air supplied to the rooms on both stories are shown in Fig. 3. The second story windows had been open during the night previous to this test from 9:15 p. m. to 7 a. m., and outdoor air had been circulated by means of the basement fan. On this test the condensing unit operated intermittently from 9:07 a. m. to 11:30 a. m., and continu-

[4] Loc. cit. Note 3.

TABLE 3—DISTRIBUTION OF DAYS CORRESPONDING TO INDICATED VALUES OF THE MEDIAN, OR AVERAGE OF THE OUTDOOR MAXIMUM AND MINIMUM TEMPERATURE

(Data based on 1683 days included between May 1 and September 30 for years 1927 to 1937, inclusive.)[a]

DAILY MEDIAN TEMPERATURE, F	40	41	42	43	44	45	46	47	48	49	50	51	52	53
NO. OF DAYS IN 11-YEAR PERIOD	1	1	0	1	2	1	5	3	9	6	10	8	18	22
AVG NO. OF DAYS PER SEASON	0.09	0.09	0	0.09	0.18	0.09	0.45	0.27	0.82	0.55	0.91	0.73	1.64	2.00

DAILY MEDIAN TEMPERATURE, F	54	55	56	57	58	59	60	61	62	63	64	65	66	67
NO. OF DAYS IN 11-YEAR PERIOD	17	14	20	22	29	28	28	36	37	38	62	44	45	47
AVG NO. OF DAYS PER SEASON	1.55	1.27	1.82	2.00	2.64	2.55	2.55	3.27	3.36	3.45	5.64	4.00	4.09	4.27

DAILY MEDIAN TEMPERATURE, F	68	69	70	71	72	73	74	75	76	77	78	79	80	81
NO. OF DAYS IN 11-YEAR PERIOD	77	55	64	67	93	81	67	81	89	72	64	46	50	53
AVG NO. OF DAYS PER SEASON	7.00	5.00	5.82	6.09	8.45	7.36	6.09	7.36	8.09	6.55	5.82	4.18	4.55	4.82

DAILY MEDIAN TEMPERATURE, F	82	83	84	85	86	87	88	89	90	91	92	93	94	95
NO. OF DAYS IN 11-YEAR PERIOD	41	26	23	18	25	5	13	6	4	2	5	1	1	0
AVG NO. OF DAYS PER SEASON	3.73	2.37	2.09	1.64	2.27	0.45	1.18	0.55	0.36	0.18	0.45	0.09	0.09	0

[a] Data from Urbana, Ill., Station of United States Weather Bureau.

ously from 11:30 a. m. to 10:07 p. m. maintaining the dry-bulb temperature between 77.0 F and 79.5 F.

At the start of the test at 9:07 a. m. the indoor relative humidity was 80 per cent and the indoor effective temperature was 75.5 deg. After the condensing unit had operated for about an hour the relative humidity decreased to 70 per cent with an accompanying effective temperature of 73.6 deg. However, during the off-period of the cooling unit both the relative humidity and effec-

tive temperature increased to values of 75 per cent and 75.3 deg respectively. This alternate decrease and increase of relative humidity and effective temperature was characteristic of the intermittent operation of the cooling plant. This characteristic was also observed in the case of the previous plants that had been

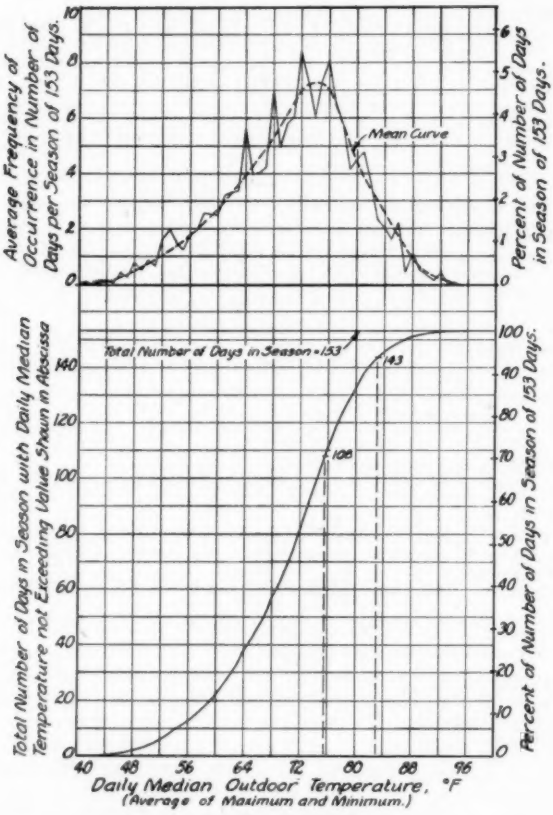
Average Frequency of Occurrence in Number of Days per Season of 153 Days.
Percent of Number of Days in Season of 153 Days.
Mean Curve
Total Number of Days in Season = 153
143
106
Total Number of Days in Season with Daily Median Temperature not Exceeding Value Shown in Abscissa
Percent of Number of Days in Season of 153 Days.
40 48 56 64 72 80 88 96
Daily Median Outdoor Temperature, °F
(Average of Maximum and Minimum.)

FIG. 5. FREQUENCY OF OCCURRENCE OF DAYS HAVING STATED DAILY MEDIAN TEMPERATURES

installed in the Research Residence. A more detailed discussion of this condition is presented in a later section of this paper. The indoor comfort conditions during the intermittent periods of plant operation were on the upper border-line for comfort and could be regarded as reasonably satisfactory.

When the plant operated continuously, as from 11:30 a. m. to 10.07 p. m., the relative humidity stabilized at about 59 per cent, representing an effective temperature of 74 deg. The indoor conditions during these periods of con-

tinuous operation could be considered as entirely satisfactory. It is of interest to note that the minimum values of the indoor relative humidity that were maintained were approximately 10 to 15 per cent greater than those maintained during the tests made in the season of 1934 with a condensing unit having a heat absorbing capacity of practically 30,000 Btu per hour. To the extent that a relative humidity of 45 per cent might possibly be considered as slightly more desirable than a relative humidity of 59 per cent, even for the same effective temperature, the indoor conditions maintained with the larger condensing unit may be considered as slightly more favorable than those maintained with the smaller condensing unit.

It may be noted that in order to maintain the same effective temperature a unit having a smaller capacity should be operated to maintain an indoor dry-bulb temperature somewhat lower than that to be maintained with a unit having a larger capacity. This fact, together with the advisability of allowing for a rise in the indoor temperature on hot days when the cooling plant was insufficient in capacity to maintain a constant temperature, made it advisable to operate the smaller plant from a room thermostat which was set at approximately 78 F rather than 80 F. The differences in the relative humidities that were obtained have been attributed partly to the difference in seasons, partly to the difference in the mean coil temperatures and partly to the difference in the depth of the coils in the two plants. A complete discussion of the observed relationship between the mean coil temperatures and the relative humidity in the house has been presented in a previous paper.[5]

Limitations in Cooling Capacity

The cooling capacity of 17,300 Btu per hour was found to be sufficient for days in which the maximum outdoor temperature rose to 95 F and the minimum outdoor temperature during the night preceding the test did not exceed 71 F. Since the summer season was relatively cool, no tests were obtainable under severe weather conditions in which the outdoor temperature exceeded 95 F. From Fig. 3 it may be observed that during the period of continuous operation the indoor dry-bulb temperature gradually rose from a value of 78.1 F at 1 p. m. to a value of 79.5 F at 6 p. m. The indoor effective temperature at 6 p. m. was 74.8 deg which was just on the border-line for comfort. It is probable that if the outdoor air had been materially hotter the rise in the indoor temperature would have been greater and the indoor effective temperature would probably have been above that represented by the upper limit of the comfort zone. In the following section an analysis is presented of the limitations of the small condensing unit when used under all weather conditions and with the method of operation employed in the Research Residence.

In a previous study[6] of the effect of methods of night operation on conditions in the Residence on the following day, it was shown that the maximum indoor temperature that was attained during the day was dependent not only on the maximum outdoor temperature, but also on the minimum outdoor tem-

[5] Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F, by A. P. Kratz, S. Konzo, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 287).

[6] Investigation of Summer Cooling in the Warm-Air Heating Research Residence, by A. P. Kratz, M. K. Fahnestock, and S. Konzo (*University of Illinois Engineering Experiment Station Bulletin*, No. 290, January 1937, pp. 73-80 and Fig. 25).

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TABLE 4—HEAT ABSORBED BY THE RESEARCH RESIDENCE

ITEM No.	ITEM	VOLUME, CU FT		DENSITY,[a] LB PER CU FT	WEIGHT, POUNDS		SPEC. HEAT[b] BTU PER LB PER DEG	TEMP. RISE IN MATERIAL,[c] DEG F	BTU ABSORBED PER DEG F RISE IN INDOOR AIR TEMP.	
		1st St.	2nd St.		1st St.	2nd St.			1st St.	2nd St.
1	2	3	4	5	6	7	8	9	10	11
a	Plaster, partition wall	75.3	62.1	103.8	7820	6450	0.24	1.00	1878	1548
b	Plaster, exposed wall	37.9	61.0	103.8	3940	6340	0.24	0.80	756	1217
c	Plaster, ceiling	52.3	55.7	103.8	5430	5780	0.24	1.00 & 0.73	1303	1012
d	Glass, interior doors	0.3	...	164.0	50	...	0.20	1.00	10	...
e	Glass, exposed	0.8	0.8	164.0	130	130	0.20	0.25	7	7
f	Studdings, partition wall	30.7	31.3	38.0	1170	1190	0.65	1.00	761	773
g	Studdings, exposed wall	29.9	41.8	38.0	1140	1590	0.65	0.64	474	661
h	Joists, ceiling	68.5	73.0	38.0	2600	2770	0.65	1.00 & 0.51	1690	918
i	Flooring	97.2	99.6	38.0	3690	3780	0.65	0.50 & 1.00	1200	2457
j	Wood, doors	20.4	15.2	28.0	570	430	0.65	1.00	370	280
k	Wood, door frames	51.8	45.9	57.0	2950	2620	0.65	1.00	1918	1702
l	Wood, window frames	36.0	35.5	38.0	1370	1350	0.65	1.00	890	878
m	Wood, baseboard	8.3	11.5	57.0	470	660	0.65	1.00	306	429
n	Wood, top moulding	5.9	1.2	57.0	340	70	0.65	1.00	221	46
o	Wood, mantel	2.0	...	57.0	110	...	0.65	1.00	72	...
p	Wood, stairs	11.2	...	38.0	430	...	0.65	1.00	280	...
q	Tile in bathroom	...	4.9	115.0	...	560	0.21	1.00	...	118
r	Plasterboard wall	...	4.3	68.0	...	290	0.40	1.00	...	116
s	Paneling	1.9	...	38.0	70	...	0.65	1.00	46	...
t	Total	...	...	...	...	...	...	...	12,182	12,162

[a] Values in column 5 for density obtained from Kent's *Mechanical Engineer's Handbook*, 10th edition, pp. 183–185.
[b] Values in column 8 for specific heats obtained from Marks' *Mechanical Engineer's Handbook*.
[c] Values in column 9 for temperature rise in material based on rise of 1.0 F in the indoor air temperature maintained until equilibrium conditions are attained.

perature that existed during the night previous to the test. For example, the maximum indoor air temperature of 81 F would result during the day, succeeding a night of operation with a given type of fan, if under upper limiting conditions a maximum outdoor temperature of 93.6 F was preceded by an outdoor minimum temperature of 62.0 F; and if under lower limiting conditions a maximum outdoor temperature of 83.6 F was preceded by an outdoor minimum temperature of 71.0 F. The same conditions could also be obtained with any maximum temperature lying between these limits when accompanied by a proper corresponding outdoor minimum temperature. It may be noted from this example that the average of the maximum and minimum outdoor temperature, or the daily median temperature, required in each case to result in the 81 F indoor air temperature was practically the same. This was also true of all

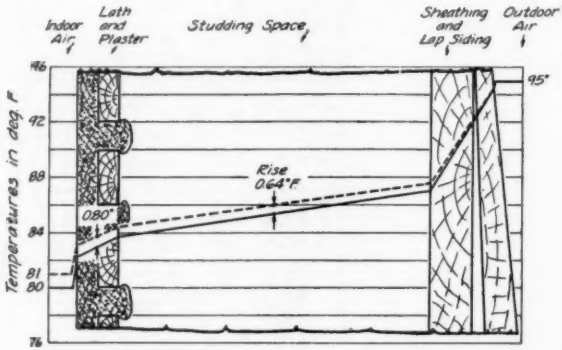


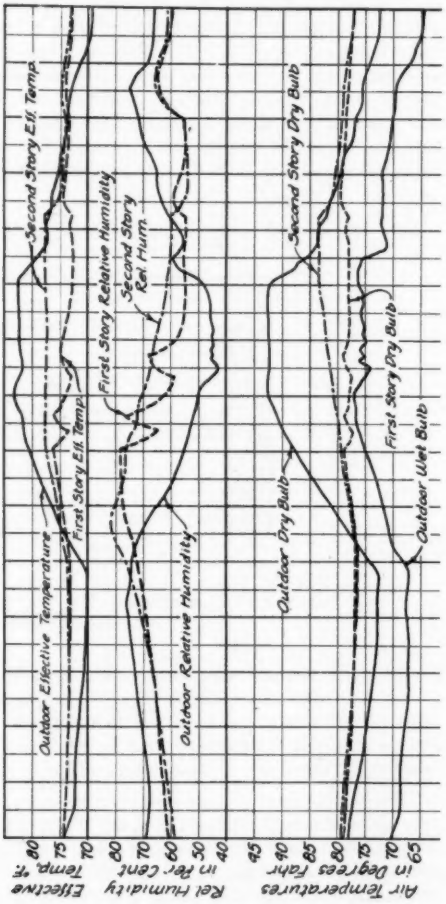
FIG. 6. TEMPERATURES OF MATERIALS COMPRISING WALL SECTION. (CALCULATED VALUES BASED ON CONDITIONS OF THERMAL EQUILIBRIUM)

of the daily median temperatures obtained from the intermediate combination of outdoor maximum and minimum temperatures required to result in this same indoor temperature. It is reasonable to expect, therefore, that the maximum indoor air temperature that is attained on a given day would bear some relation to the daily median temperature.

That such a relationship exists is shown by the data plotted in Fig. 4. These data were obtained during the summer of 1937 and show the maximum indoor temperatures that were obtained when cooling was accomplished only by the use of outdoor air at night, and the condensing unit was not operated. The points representing these data are shown as circles in Fig. 4 and a full line is shown drawn through the points. It may be observed that for a daily median temperature of 75.5 F the maximum indoor temperature was 79.0 F.

The points representing the data obtained for the maximum indoor temperature when the small-sized condensing unit was operated are indicated by crosses, and a broken line drawn through the points represents the average. The intersection of the broken line with the full line is of particular significance as representing a critical daily median temperature, since it shows that for all

values of the daily median temperature exceeding 75.5 F some amount of artificial cooling was required, and that for all values less than 75.5 F cooling with outdoor air at night alone was sufficient to maintain suitable indoor con-



ditions. For example, for a daily median temperature of 83 F, if night air cooling alone was used, the indoor temperature would have risen to approximately 84 F; whereas if the small-sized condensing unit had been operated with the room thermostat set at 78 F, the indoor temperature would not have exceeded 80 F.

The daily median temperature of 83 F, which was exceeded only once dur-

ing the cooling season of 1937, may be considered as the probable upper limit for the range of application of the small-sized condensing unit. It is most probable that for daily median temperatures exceeding 83 F, the indoor tem-

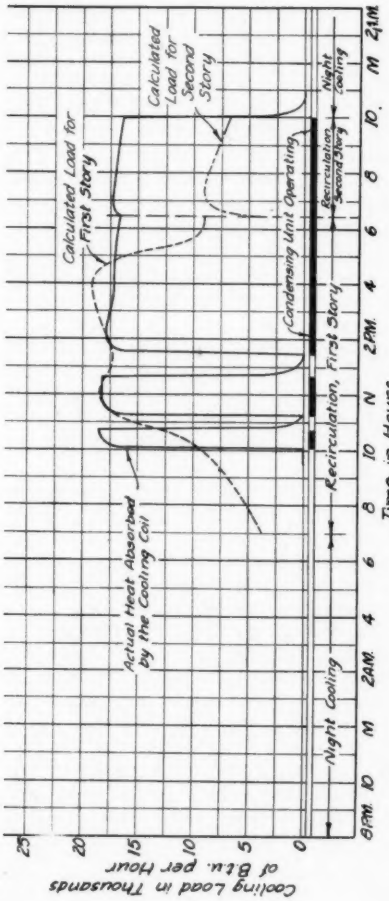


FIG. 7. ACTUAL AND CALCULATED COOLING LOAD ON RESIDENCE AND AIR TEMPERATURES. TEST NO. 19, SERIES 2-37, AUGUST 17, 1937

peratures would have risen above 80 F and the comfort conditions would not have been entirely satisfactory during the entire day. This is indicated by the fact that the broken line in Fig. 4 intersects the 80 F maximum indoor temperature line at a point corresponding to a daily media temperature of approximately 83 F. However, even with these higher outdoor temperatures, the resulting indoor conditions would have been much more acceptable than if

no cooling had been used. Additional data under extreme weather conditions are desirable in order to investigate the actual resulting indoor comfort conditions.

From a statistical study of the daily median temperatures for a given season in Urbana, Ill., the number of days on which satisfactory indoor conditions would result, either from the use of air from outdoors at night or by supplementing such cooling with the use of the small condensing unit, can be determined by employing the data shown in Fig. 4. Such a study has been applied to a season represented by the average of 11 seasons extending over the years of 1927 to 1937 inclusive and the results are shown in Table 3. This table shows a summary of the total number of days during the 11-year period for which the daily median temperatures assumed the values indicated. The data shown are for the total 1683 days included in all of the seasons between May 1 and September 30 (153 days per season) for the 11 years, and represent mild as well as severe summers. It may be noted that a daily median temperature of 75.5 F was obtained about 85 times during the 11-year period, averaging about eight days per season; whereas a daily median temperature of 83.0 F occurred 26 times during the 11-year period, or an average seasonal frequency of 2.4 days.

The data in Table 3, showing the average number of days during a season on which a given daily median temperature was attained, have been represented graphically in the upper part of Fig. 5. The broken line represents a mean of the points plotted. The values indicated on the right hand ordinate show the frequency in terms of percentage of the number of days in a season comprised of 153 days.

The lower curve in Fig. 5, representing the total number of days in the season having daily median temperatures not exceeding the value shown on the abscissa, was obtained by integrating the area under the curve included between the vertical ordinates of the origin and the stated value in the upper part of Fig. 5. It may be noted, for example, that for approximately 108 days in the total season of 153 days, the daily median temperature would be less than 75.5 F. The latter was indicated in Fig. 4 as representing a critical or limiting daily median temperature. Hence Fig. 5 indicates that cooling with air from outdoors at night alone would be sufficient to maintain comfortable conditions in the Research Residence for 108 days, or 70.6 per cent of the average season of 153 days; but would not be adequate for the remaining 45 days.

Similarly Fig. 5 indicates that for approximately 143 days in the season, the daily median temperature would not exceed the 83.0 F shown in Fig. 4 as the upper limit for the small condensing unit. That is, the small condensing unit, when used in connection with cooling with outdoor air at night by means of the basement fan, would be sufficient to maintain desirable comfort conditions in the Residence on 143 days, or 93.4 per cent of the season, but would not be sufficient in capacity to prevent the indoor temperature from exceeding 80 F on 10 days. These data represent a typical average season only, and considerable deviations from the average may occur during any given season. For example, during the relatively mild season of 1937 the daily median temperature exceeded 83.0 F on only one day; whereas during the extremely severe summer season of 1936, the daily median temperature exceeded 83.0 F on 28 days. Furthermore, the limitations shown in Fig. 4 apply only to the method of operation and to the operating conditions as main-

tained in the Research Residence. If a larger condensing unit had been used, the limiting value of the daily median temperature, as shown in Fig. 4, would have been greater than 83.0 F and, as indicated by Fig. 5, the unit would have had sufficient capacity to maintain acceptable conditions for more than 143 days per season.

The data presented indicate the feasibility of installing refrigerating equipment of small capacity in residences, particularly in those cases in which the initial cost of installation is a more important factor than the maintenance of ideal indoor conditions in the most severe weather. The data furthermore show the limitations resulting from the use of such a small condensing unit when operated under the conditions maintained in the Research Residence.

Heat Absorbed by the Structure

It was observed in connection with Fig. 3 that during the period of continuous plant operation, when the actual heat absorbed by the cooling coil was only 65 per cent of the maximum calculated load on the Residence, the indoor dry-bulb temperature rose from 78.1 to 79.5 F, or a total rise of 1.4 F. During the season of 1936 a similar rise in indoor temperature was also observed. For example, on July 7, 1936 [7] when the outdoor maximum temperature was 104.0 F, the calculated cooling load reached a maximum value of 37,000 Btu per hour, the actual heat absorbed by the cooling coil was 24,000 Btu per hour, and the indoor air temperature rose from 78.5 to 81.5 F, or a total rise of 3.0 F. In both examples, the cooling plant in use did not absorb heat at the rate indicated by the maximum calculated value of the cooling load. In view of the high rate at which the calculated cooling load increased as compared with the actual heat absorption, the magnitude of the rise in the indoor air temperature that was actually obtained appeared to be extremely small. The fact that this rise in the indoor air temperature was not greater has been attributed to heat absorbed by the building structure and furniture and not in turn given up to the indoor air.

The approximate amount of heat that would be absorbed by the Residence structure if the indoor air temperature was allowed to increase 1.0 F and equilibrium was assumed to be established under these new conditions, was obtained as shown in Table 4 and Fig. 6. The weights of the different materials (columns 6 and 7) comprising the structure were calculated from the space occupied by each material, as determined by actual measurement, (columns 3 and 4) and the density of the material (column 5). The temperature rise (column 9) for those materials which were subjected to a 1.0 F rise in indoor air temperature on both sides of the material, such as a partition wall, was assumed to be also 1.0 F.

Owing to the existence of a temperature gradient, the temperature rise in those materials which were exposed to a 1.0 F rise in indoor air on one side and to a constant temperature on the other side, as in the case of an exposed wall, would be less than 1.0 F. The magnitudes of these temperature rises, based on the assumption that the indoor air temperature was allowed to increase 1.0 F and that equilibrium was established under the new conditions, were obtained by calculating the temperature gradient through the wall, making use of the general equations of heat flow and the accepted values of the conduc-

[7] Loc. cit. Note 5.

tivities of the material. In the case of an exposed wall, calculations for the temperature gradient, and hence the surface temperatures, were based first on an indoor air temperature of 80 F and an outdoor air temperature of 95 F. The average of the two surface temperatures of the lath and plaster was then considered as the temperature of the slab comprised of the lath and plaster, and the average of the surface temperature of the sheathing and lap siding was considered as the temperature of the slab composed of these materials.

TABLE 5—SUMMARY OF RESULTS OF TESTS FOR SEASON OF 1937

WEATHER DATA	
1. Total hours above 85 F for season of 1937	301.9
2. Total hours above 90 F for season of 1937	40.1
3. Total degree-hours above 85 F for season of 1937	862.4
4. Total degree-hours above 90 F for season of 1937	63.7
NIGHT AIR COOLING	
5. Total number of nights with cooling by fan	76
6. Total running time for fan, hours	845.9
7. Average rate of power input to fan, watts	453
8. Total power input to fan, kilowatt-hours	390.2
AIR RECIRCULATION WITHOUT COOLING	
9. Total number of days	75
10. Total running time for fan, hours	674.3
11. Average rate of power input to fan, watts	432
12. Total power input to fan, kilowatt-hours	291.4
AIR RECIRCULATION WITH COOLING	
13. Total number of tests	28
14. Total running time for fan, hours	220.6
15. Average rate of power input to fan, watts	437
16. Total power input to fan, kilowatt-hours	97.4
COMPRESSOR OPERATION	
17. Total number of hours	199.0
18. Average rate of power input, watts	1396
19. Total power input, kilowatt-hours	277.9
20. Total quantity of cooling water, gallons	11,961
COMBINED TOTALS	
21. Total running time for fan including night air cooling, hours	1756.8
22. Total power input to fan including night air cooling, kilowatt hours	779.0
23. Total power input to compressor and fan including night air cooling, kilowatt-hours	1056.9

Similar calculations were then made for an indoor air temperature of 81 F and an outdoor air temperature of 95 F. The difference in the temperatures of each slab of material obtained by these two calculations was considered as the temperature rise for that material. A graphical representation of the temperatures established in an exposed wall, for values of the indoor air temperature of 80 F and 81 F, is shown in Fig. 6. It may be noted that for an increase in the indoor air temperature of 1.0 F, the increases in temperature for the lath and plaster and for the studs were 0.80 F and 0.64 F respectively. The increases in temperature for the sheathing and lap siding were negligibly small. Similar calculated values for other materials are shown in column 9 of Table 4.

The total amounts shown in columns 10 and 11 in Table 4 indicate that the heat absorbed or given up by the Research Residence was 12,182 Btu on the first story and 12,162 Btu on the second story, or a total amount of 24,344 Btu for each one degree change in indoor air temperature. These values are for the building structure alone and do not include an allowance for the heat that could be absorbed by the furniture, floor coverings, light fixtures, books, and equipment. Therefore, an actual change of 1.0 F in the indoor air temperature is representative of a very large heat exchange. If the indoor air temperature were reduced 1.0 F a large amount of heat would be given up to the air by the structure and would represent a large additional load on the plant. Conversely, if the indoor air temperature rose 1.0 F the heat absorbed by the walls would tend to relieve the plant. For example, if the indoor air temperature changes 1.5 F during a four-hour period, the rate at which heat is being absorbed or given up by the structure will be greater than 9000 Btu per hour; which is equivalent to ¾ ton of refrigerating capacity during the four-hour period.

These results emphasize the necessity, particularly where the plant capacity is limited, of starting the cooling plant each day before the indoor temperature has risen above that required for comfort conditions. If the indoor temperature is allowed to rise above 80 F, a comparatively high capacity will be required, even on a mild day, in order to effect a rapid decrease in the air temperatures, particularly at a time when the load on the cooling plant is increasing. Although these results apply to a larger extent to residence conditions, where the occupancy load is small, the effect of heat exchange between the air and the structure should also receive consideration in the case of other types of structures and under other operating conditions, particularly in installations in which a rapid reduction in the indoor air temperature is desired.

RESULTS OF TESTS WITH COOLING ON FIRST STORY DURING THE DAY AND ON SECOND STORY AT NIGHT

General Conditions

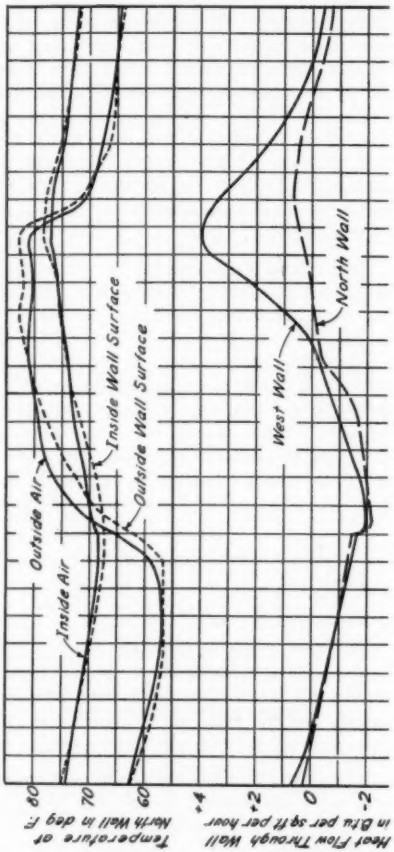
The test made on August 17, 1937, was selected as typical of the operation obtained when cooled air was distributed to the first story rooms only until 6:30 p. m., and to the second story rooms only after 6:30 p. m. The results for the tests, made under conditions of operation for Series 2-37, are shown in the second and third columns of Table 1 and in Fig. 7.

The values listed in the second column of Table 1 show the average conditions that were maintained in the first story rooms at 3:30 p. m.; while the values listed in the third column show the average conditions that were maintained in the second story rooms at 7:30 p. m. The outdoor air temperature was about 93 F for a period of four hours and the day may be considered as a medium warm day.

Operating Characteristics

The operating characteristics for this method of operation are shown in Fig. 7. Outdoor air had been circulated through all the rooms of the house during the night previous to the test, from 8 p. m. to 7 a. m., by means of the basement fan. At 7 a. m. the recirculation of the indoor air, together with the

addition of outdoor air for the purpose of ventilation, was started on the first story alone. Practically the same quantities of both recirculated air and ventilation air, as were circulated on both stories in the first series of tests, were circulated on each individual story in this second series of tests. Hence the



quantities of air circulated, based on one story alone, were approximately equivalent to six recirculations per hour and two changes of outdoor air per hour, making a total of eight air changes per hour delivered by the fan. The ratio of the heat brought in by ventilation air to the total heat gain for the story was greater in this series than it was in the first series of tests.

On the test shown in Fig. 7 the condensing unit operated intermittently from 10:05 a. m. to 1:31 p. m. As in previous tests, the intermittent periods of

plant operation were accompanied by large changes in the relative humidity and in the effective temperature. The comfort conditions during the off-periods of the plant were not as satisfactory as those maintained during the on-periods of the plant. These results are consistent with all of the results

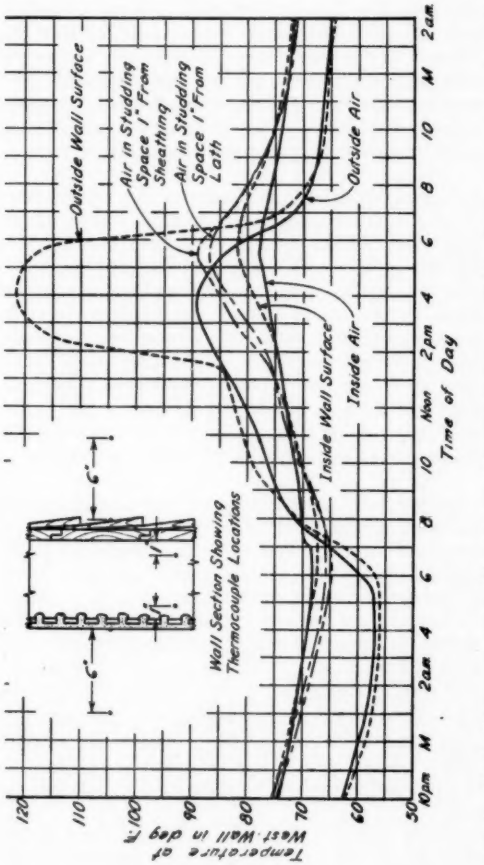

Outside Wall Surface
Air in Studding Space 1" From Sheathing
Air in Studding Space 1" From Lath
Outside Air
Inside Wall Surface
Inside Air
Time of Day
10pm M 2am 4 6 8 10 Noon 2pm 4 6 8 10 M 2am
Temperature of West Wall in deg F
120 110 100 90 80 70 60 50
Wall Section Showing Thermocouple Locations
6"
1"
6"

FIG. 8. HEAT FLOW THROUGH WALLS AND TEMPERATURES AT WALL SECTIONS FOR AUGUST 13, 1937

of tests made over the period of from 1934 to 1936, inasmuch as they all indicated that the indoor comfort conditions were more satisfactory on relatively hot days when the plant operated continuously than they were on the milder days when the plant operated intermittently. In this connection it may be mentioned that the use of a cooling unit or operating method which was just sufficient to maintain the desired indoor temperature at all times would eliminate these off-periods and the accompanying periods of high humidity. Any

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method of control or operation which would tend to modulate the capacity of the cooling unit to more nearly conform with the cooling requirements should prove more satisfactory than the intermittent operation of a unit having fixed capacity.

On this test, the condensing unit operated continuously from 1:31 p. m. to 10 p. m. The indoor comfort conditions on the first story remained constant and were entirely satisfactory until 6:30 p. m. From 6:30 p. m. until 10 p. m., when the cooled air was distributed to the second story rooms only, the indoor temperature, humidity, and effective temperature on the first story increased slightly, but the conditions were acceptable. It is probable that on extremely severe days, the conditions on the first story after 6:30 p. m. might prove uncomfortable, and an additional investigation under these severe weather conditions is required.

It may be observed that the temperatures in the rooms on the second story increased steadily from 76.8 F at 7 a. m. to 83.5 F at 6:30 p. m. Since no air was recirculated on the second story during this interval, and the windows remained closed, the air conditions in the second story rooms were not satisfactory. The contrast between the warm, stuffy environment on the second story and the cool, fresh environment on the first story was particularly striking. The conditions were so noticeably different, particularly in late afternoon, that the occupants of the house preferred to remain on the first story. This limitation and restriction in the usage and occupancy of the entire house may prove to be undesirable in some installations, where sewing rooms and utility rooms are located on the second story. The normal living habits of the occupants should, therefore, be considered in the application of zoned installations of this nature.

During the interval when cooling was applied only on the first story, the relative humidity on the second story decreased from 77 per cent at 10 a. m. to 60 per cent at 6:30 p. m., whereas ordinarily when no cooling was done in the house the indoor relative humidity tended to remain fairly constant. This reduction in relative humidity on the second story was caused by a reduction in the moisture content of the air and not merely by the higher air temperature. The moisture content decreased from 116 grains per pound of dry air at 10 a. m. to 104 grains per pound at 6:30 p. m. This reduction of 12 grains per pound of dry air on the second story accompanied a reduction in the moisture content on the first story amounting to 37 grains per pound, and indicated that the circulation of relatively cool dry air on the first story appreciably affected the conditions on the second story. As a result of this the effective temperature on the second story remained practically constant in spite of the fact that the dry-bulb temperature increased.

At 6:30 p. m. when the cooled air was distributed only through the registers on the second story, the reduction in relative humidity and temperature occurred quite rapidly during the first hour. The reduction in temperature occurred very slowly after that time due to the effect of the heat absorbed in the structure. This latter effect is also indicated, as shown in the bottom part of Fig. 7, by the wide discrepancy between the calculated cooling load on the second story and the actual load absorbed by the plant. During the interval from 8 p. m. to 10 p. m. the actual load was more than twice as great as the calculated load, while the indoor air temperature on the second story decreased from 80.5 F to 79.3 F, giving a total drop of only 1.2 F. The degree of com-

fort on the second story, as measured by the effective temperature, did not attain that represented by 74.5 deg until approximately two hours after the cooling was begun on the second story. Even at 10 p. m., 3½ hours after the cooling was begun on the second story, the effective temperature was 74 deg, only slightly below the upper border line of comfort. It is most probable that on a hotter day the conditions on the second story would not be completely satisfactory between the hours of 6:30 p. m. and 10 p. m. Additional data on this point are desirable.

In general, therefore, subject to the limitations discussed, the method of cooling on the first story during the day and on the second story during the night may be considered as reasonably satisfactory. The method requires a zoned duct system in which all of the rooms on one story are supplied with

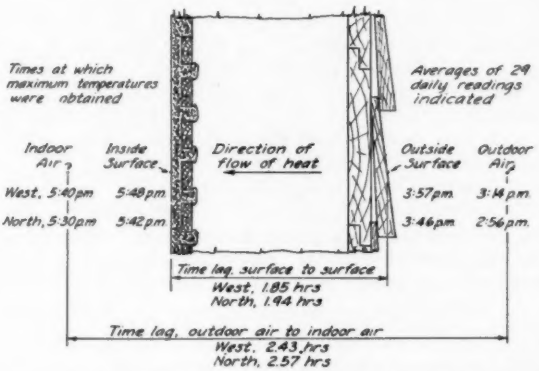

FIG. 9. AVERAGE VALUES OF TIMES AT WHICH MAXIMUM TEMPERATURES WERE ATTAINED IN WALL SECTIONS

air from a trunk duct separate from those supplying air to other stories. Such an arrangement may not be practical in all installations. The restriction in the usage of the house may prove objectionable in some cases. Furthermore, the same unit when used to cool the entire house during the comparatively mild season of 1937 proved almost as satisfactory without restricting the space suitable for occupancy. On the whole, the comfort conditions attained on the first story rooms were slightly more satisfactory than those maintained when the small condensing unit was used to cool the entire house. However, the comfort conditions maintained during the off-periods of the plant were subject to improvement, and furthermore more satisfactory conditions as a whole would have resulted if the number of off-periods of the plant could have been reduced. The conditions attained on the second story were just on the upper border line of comfort, and a larger capacity would have been required in order to produce a more rapid reduction in the temperature.

Summary of Seasonal Results

Table 5 gives a summary of the total quantities obtained during the summer of 1937. This season was comparatively cool and in this respect was in decided

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contrast to the preceding hot summer. The total of 862 degree-hours above 85 F, which was about one-fifth that obtained for 1936, is a good indication of the difference between the two seasons. The total numbers of days with temperatures 85 F or over and 90 F or over for the season of 1937 were 61 and 23 respectively. The severity of the season, as measured by degree-hours, is apparently determined largely by the total number of days on which the maximum temperature exceeds 90 F. The outdoor humidities were more nearly normal during 1937, and were much higher than those obtained during the hot season of 1936.

From item (21) in Table 5 it may be observed that the total time that the fan was running, including cooling with outdoor air at night and recirculation during the day, was 1756.8 hours. The total cost for electricity for operating the fan, based on a rate of 3.1 cents per kilowatt-hour for electricity, was $24.13. The total time of operation of the compressor was 199.0 hours and the cost of electricity for this purpose was $8.61. The total amount of water actually used for the condensing unit was 11,961 gal. At the prevailing rate of 33 cents per thousand gallons, the total cost for water was $3.95. The cost for both electricity and water for the season was $36.69. Even allowing for two days in early June during which the compressor was not ready for operation, the total seasonal cost would not have exceeded $40. It may be noted that the cost of operation of the condensing unit was only one-third of the total cost, the cost of operation of the circulating fan accounting for the other two-thirds. Since the variations in seasonal demand were extremely wide, a comparison of the cost obtained with the small condensing unit with that obtained with other types of cooling units might to a certain extent be misleading.

RESULTS OF STUDY WITH HEAT-FLOW METERS

Temperature Studies

The data obtained from the two heat-flow meter installations over a period of 29 days were plotted on a continuous chart, and this chart was used as a basis for the analysis of results. A typical section from the chart for a 28-hour period is shown in Fig. 8. It may be noted that the temperatures were continually changing and that thermal equilibrium was never attained. This cyclical variation both in the temperatures and in the direction of heat flow does not conform with the assumption of thermal equilibrium, which is invariably made in the calculation of heat losses or gains under given conditions of sun effect and temperature difference between the indoors and outdoors. The top set of curves shows the air and surface temperatures obtained at the north wall section, while the lower set shows those obtained at the west wall section. The two curves in Fig. 8 between the upper and lower sets show the actual amounts of heat passing through the inside surface of the two walls, as indicated by the readings of the heat-flow meters. These meters were located one on each wall, directly below the thermocouples installed to obtain the temperature gradient through the wall.

It may be observed from Fig. 8 that there existed an appreciable interval between the time at which the outdoor air temperature attained a maximum value and that at which the indoor air temperature reached a maximum. In order to obtain the average values of the times at which various parts of the

wall sections attained their maximum temperatures the data obtained from the 29 days were averaged, and are shown in Fig. 9. The values for the west wall, which was exposed to the sun, were not appreciably different from those for the north wall, which was not exposed to the sun. The lag between the times at which the outdoor air temperature was a maximum and the indoor air temperature was a maximum was approximately 2.5 hours. Also, the lag between the times at which the outdoor surface temperature reached a maximum and indoor surface temperature attained a maximum was approximately two hours.

Studies of Heat Transmission

Preliminary calculations of the conductance of the wall section, as determined from actual measurements of the surface temperatures and the heat flow through the heat-flow meters, gave results that were materially less than the values obtained for similar walls from laboratory determinations under conditions of thermal equilibrium. This indicated either that the conductance value determined from laboratory investigations was in error or that conditions of thermal equilibrium were not attained in the actual wall exposed to normal temperature variations. The former supposition is not tenable, whereas the latter condition was actually known to exist.

This discrepancy between the measured and the calculated values of the heat flow through the wall is illustrated by the curves in Fig. 10. The calculated values were determined from the following equation [8]:

$$H = \frac{UeI}{f_o} + U(t_o - t_i)$$

in which

H = heat transmitted through wall in Btu per square foot per hour,
U = overall coefficient of heat transfer, Btu per square foot per hour per degree difference in temperature,
e = coefficient of absorption of solar radiation, (0.40 for weathered white paint),
I = solar intensity normal to the surface, in Btu per square foot per hour,
f_o = outside surface coefficient, Btu per square foot per hour per degree Fahrenheit,
t_o = outdoor air temperature, degrees Fahrenheit,
t_i = indoor air temperature, degrees Fahrenheit.

It may be noted from Fig. 10 that the maximum value of the calculated heat flow through the west wall was equal to 6.3 Btu per square foot per hour and occurred at 4 p. m.; whereas the maximum value of the heat actually transmitted into the room as measured by the heat-flow meter, was 3.9 Btu per square foot per hour and occurred at 6 p. m. The time lag between these maximum values was 2 hours.

It was also observed in these studies that at certain times of the day the temperature in the studding space was higher than those of both the indoor and outdoor surfaces of the wall. That is, under these conditions the heat flow in the wall was occurring simultaneously to the outdoors and to the indoors. The calculated values of heat transmission are of little significance under such conditions; the significant values being given by the heat-flow meter, which on account of its location on the inside surface of the wall measures the actual flow of heat into the room.

During periods when the outdoor air temperature was increasing, the temperature of the material comprising the wall structure was usually increasing

[8] Loc. cit. Note 6. Appendix B. Method Used for Calculation of Cooling Load, pp. 131-134.

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also. As explained in the section on heat absorbed by the structure, the total amount of heat impressed on the wall appears not only as heat passing through the heat-flow meter but also as heat which tends to increase the temperature of the wall structure, and the discrepancy between the maximum values shown in Fig. 10 has been attributed to this factor, although it is also possible that the presence of the heat meter itself may have a disturbing effect. It is evident, however, that in walls of the type tested, the lack of equilibrium conditions results in changes in the amount of heat absorbed or given up by the

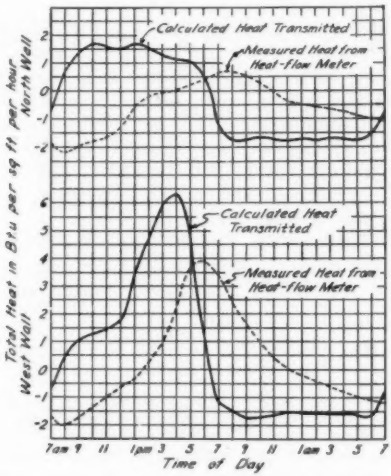
Calculated Heat Transmitted
Measured Heat from Heat-Flow Meter
Total Heat in Btu per sq ft per hour
North Wall
Calculated Heat Transmitted
Measured Heat from Heat-Flow Meter
West Wall
7am 9 11 1pm 3 5 7 9 11 1am 3 5 7
Time of Day

FIG. 10. HEAT FLOW THROUGH WALLS OF RESEARCH RESIDENCE FOR AUGUST 13, 1937

structure and affects the amount of heat transmitted through the wall; so that the calculated values of the heat flowing through the wall at any given time, when used as an index of the heat actually transmitted to the air in the structure, are not only meaningless, but may also be misleading. Furthermore, under summer conditions the maximum amount of heat transmitted through the inside surface of the wall into the inside of the structure may be less than the maximum heat flow through the wall as calculated from the indoor and outdoor conditions.

SUMMARY AND CONCLUSIONS

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

(1) In order to maintain the same effective temperature, the unit having the smaller capacity was operated to maintain an indoor dry-bulb temperature somewhat lower than that maintained with a unit having a larger capacity. Furthermore, it was

advisable to allow for a rise in the indoor temperature on hot days when the cooling plant did not have sufficient capacity to maintain a constant temperature.

(2) The cooling capacity of 17,300 Btu per hour was found to be sufficient for both stories for days on which the maximum outdoor temperature rose to 95 F and the minimum outdoor temperature during the night preceding the test did not exceed 71 F. The daily median temperature (average of maximum and minimum outdoor temperature) of 83 F may be considered as the upper limit of the range of application of the small-sized condensing unit.

(3) A statistical study of the daily median temperatures during a given number of seasons was made, and from it the limitations of the application of the small condensing unit were determined.

(4) Subject to limitations over a small portion of the cooling season, the use of a condensing unit of small capacity was shown to be feasible for residences, especially in those cases in which the initial cost of installation is a more important factor than the maintenance of ideal indoor conditions.

(5) The fact that the rise in indoor air temperature accompanying the operation of the small condensing unit on a warm day was small was attributed to the heat absorbed by the building structure and furniture.

(6) The calculations for the heat absorbed by the structure of the Research Residence indicated that a rise of 1.0 F in the indoor air temperature was equivalent to a heat absorption of over 24,000 Btu.

(7) A comparatively large plant would probably be required, even on a mild day, in order to effect a rapid decrease in the indoor air temperatures, particularly if the load on the cooling plant were increasing.

(8) The off-periods occurring during periods of intermittent operation of the plant were accompanied by comparatively large changes in the relative humidity and in the effective temperature.

(9) Subject to the limitation of restricted occupancy, the method of cooling on the first story during the day and on the second story during the night may be considered as reasonably satisfactory.

(10) The studies made with heat-flow meters applied to the walls indicated that considerable discrepancy existed between the measured values and the calculated values of the heat flow through the wall. Furthermore, thermal equilibrium was never attained in the wall section under actual conditions.

(11) Except under equilibrium conditions, the total amount of heat impressed on the wall appears not only as heat transmitted to the indoor air but also as heat serving to increase the temperature of the wall structure. The discrepancy between the maximum amount of heat actually transmitted to the indoor air, as measured by the heat-flow meter, and the maximum calculated heat transmission for the wall has been attributed to heat absorbed by the wall structure.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the summer cooling investigation (1937) in the Research Residence[9] at the Uni-

[9] The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

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versity of Illinois conducted by the Engineering Experiment Station of which M. L. Enger, dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to R. B. Engdahl, research graduate assistant, and to R. B. Knight, special research graduate assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

No. 1096

HEAT TRANSFER THROUGH SINGLE AND DOUBLE GLAZING

By M. L. Carr,* R. A. Miller,** Leighton Orr† (*MEMBERS*), and Alan C. Byers *** (*NON-MEMBER*)
Pittsburgh, Pa.

IN consideration of the need for data on the relationship of windows and their appurtenances to indoor atmospheric environment which has developed through the increased use of air conditioning, the chairman of the A.S.H.V.E. Committee on Research during 1936 appointed a Technical Advisory Committee on Air Conditioning Requirements of Glass, consisting of M. L. Carr, chairman, R. A. Miller, vice-chairman, F. L. Bishop, A. N. Finn, E. H. Hobbie, R. J. Lillibridge, F. W. Parkinson, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker, G. B. Watkins and F. C. Weinert.

In keeping with a program adopted by this Committee, the Research Laboratory is to make available the results of research carried on by manufacturers or others. This paper represents an analysis and certain deductions made by the Laboratory of data collected, prior to the organization of the Committee, by Pittsburgh Testing Laboratory for the Pittsburgh Plate Glass Co.

It is believed by Col. W. A. Danielson, former chairman of the Committee on Research, that research of a similar character can be carried out under a cooperative plan by the Research Laboratory and research organizations in allied industry, which may greatly expand the useful results of the Society's research.

In accordance with this policy, the Research Laboratory may be called upon to work closely with a manufacturer or anyone else carrying on research, by suggesting a method of attack and procedure for carrying on the work and, if found desirable, to cooperate in the collection and analysis of the data with a view that the Laboratory may use such information in the publication of technical papers.

THE rapid increase in air conditioning applications, together with the demand for high moisture content in the indoor atmosphere, has centered attention on the window as an important source of heat loss by transmission and infiltration. Greater attention has also been directed to such other important effects as condensation and frosting of the inside glass surface.

The transmission of heat through windows forms an important part of the heat loss or gain from or to winter and summer air conditioned space. The values in use by the profession and included for a number of years in THE A.S.H.V.E. GUIDE are probably based upon the results of different investigators. The values contained in the 1937 edition give transmission values of 1.13, 0.45 and 0.28 Btu per square foot per hour per degree temperature dif-

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** Technical Sales Engr., Pittsburgh Plate Glass Co.
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*** Research Engr., A.S.H.V.E. Research Laboratory.

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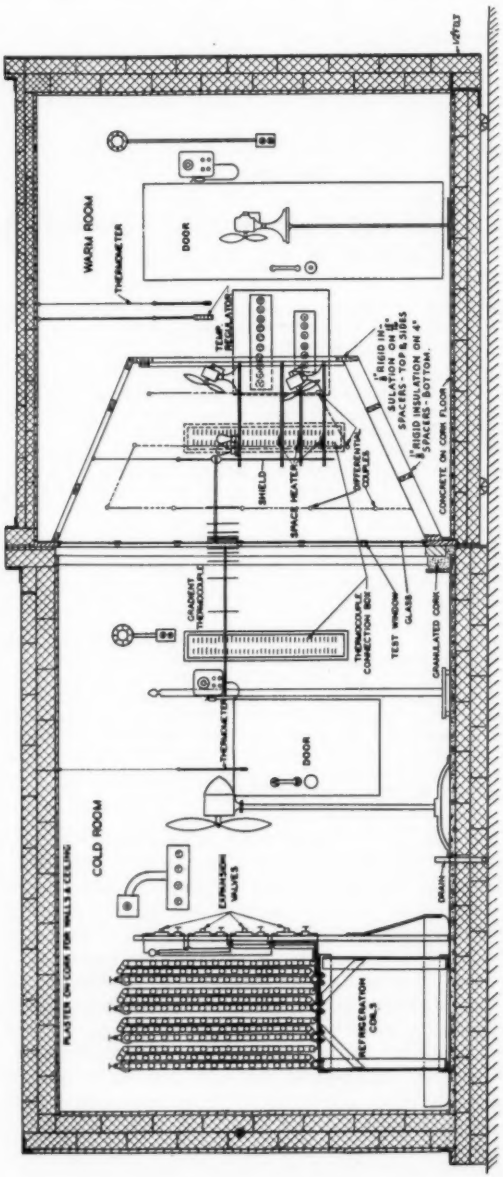

FIG. 1. VERTICAL CROSS-SECTION OF THE TEST EQUIPMENT WITH THE 36-LIGHT WINDOW INSTALLED

ference, for single, double and triple glazed windows, respectively. No differentiation is made in these coefficients depending upon the size of the window panes, the method of installation, size and shape of the muntins separating the individual panes, and other important factors. Neither is there any differentiation in the heat transmission for multiple glazing with different air spaces, with different air velocities, and with varying mean temperature between the inside and outside air. This paper gives information on a number of these points.

APPARATUS

A cross-section of the heat flow apparatus used in these tests is shown in Fig. 1. It consists of a Cold Room, in which outdoor air conditions are simulated, and a Warm Room in which indoor air conditions are reproduced, and which contains the guarded enclosure or hot box. The test specimen forms the dividing partition between these two rooms.

The Cold Room is an enclosure about 8 ft by 11 ft by 8 ft high. Its walls, with the exception of the test specimen, and its floor and ceiling are made of 8-in. cork, plastered on the inside. A cork insulated vestibule with refrigerator doors provides access when the open end is closed by the test panel.

The cooling system consists of four independent motor-driven compressor units, each connected to its own cooling coil located in the Cold Room. The capacity of this system is sufficient to maintain a Cold Room temperature of −30 F, with the test panel and a Warm Room temperature of 70 F.

The Warm Room, about 8 ft by 9 ft by 8 ft high, is of similar construction and is mounted on casters to permit its removal from the Cold Room while a test specimen is being installed. A flexible system of electrical heating is provided. It is controlled either automatically with a mercurial thermostat, or manually as desired. Both the Warm Room and the Cold Room are provided with fans for air circulation, electric lights, electric outlets, both 110 and 220 volts, connections for thermocouples and telephones.

The Guarded Enclosure is shaped like the frustum of a four-sided pyramid. Its larger end is open and of the same size as the windows to be tested, and is fastened with a calked air-tight joint to the wood frame in which the test specimen is installed. The walls are of double insulated construction, and contain about 50 differential thermocouples threaded through the walls, with approximately one couple in each 2 sq ft of the inside and outside wall surface. All the differential couples on any one wall of the Guarded Enclosure are in series. Provision is also made to connect all five of these groups in series. The Guarded Enclosure is provided with its own set of heaters, shielded from the test panel, fans and lights. The electric input to all electrical appliances in the Guarded Enclosure is metered.

Temperature measuring apparatus consists of 48 copper-constantan thermocouples, 24 in the Cold Room, 24 in the Warm Room and Guarded Enclosure. Most of the couples are of No. 22 gage B and S, a few are of No. 30 gage. Terminal boxes for the thermocouples are permanently installed in the Warm and Cold Rooms. Cables lead from these boxes to selector switches on the instrument bench. The thermocouple wires and the potentiometer used with them are calibrated against a precision thermometer. The electrical metering apparatus is an integrating watt-hour meter which could be read to one watt-hour. An indicating wattmeter is used as a check. Light threads are used

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to determine the location and direction of air movement, while a Kata thermometer is used to measure the velocities.

Humidity is measured by a mercurial thermometer type of hygrometer and by a set of wet and dry junction thermocouples placed in the air current from a fan. Some control of the humidity is provided by means of silica gel.

TEST SPECIMENS

Two test panels were used in these tests. They were identical in over-all dimensions, being made for a window opening 7 ft, 4⅜ in. square. They differed only in the size and number of lights. One was for nine lights, 28 13/16-in. square, with 3/16 to ¼ in. bearing on all sides; the second window had 36 lights, 14 5/16-in. square, with practically the same bearing. Glazing was on the outside. Each sash was installed in the apparatus and then glazed with ⅛-in. (nominal) plate glass, set in ordinary sash putty. Fig. 2 shows the

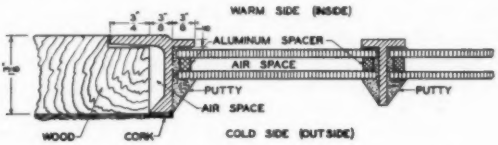

FIG. 2. DETAIL OF TEST PANEL—UPPER SIDE IN CONTACT WITH WARM AIR

window structure tested. Aluminum spacer strips, 3/16-in. wide, having the thickness of the air space desired, were placed between the lights in the double glazed sash.

TEST PROCEDURE

To obtain a desired temperature in the Cold Room, the number of compressors operating and the back pressure of each were adjusted in the light of previous experience. Regulation of the temperature was accomplished by manual manipulation of the back pressure by the regulating valves.

The watt-hour meter was read when thermal equilibrium was established, at half-hour intervals throughout the five-hour duration of the test, and at the end, while temperatures were measured six to eight times during the test. The Warm Room was held at approximately 70 F, and the Cold Room at some definite temperature between −30 F and 60 F. In a series of tests on any sash arrangement the coldest test was run first. No extreme care was taken to attain balanced conditions at some definite temperature, as it was considered more important to adjust the refrigerating equipment to obtain a temperature somewhere near the one desired and then to allow conditions to reach a balance undisturbed by change of regulating valves.

Temperatures were measured at points opposite the center of each light, 1-in. from the glass surface on both the warm and cold sides in the case of the nine-light window. Temperatures for plotting gradients were measured on a line perpendicular to the center of the window, with the thermocouples located in contact with the glass and spaced ⅛, ¼, ½, 1, 2, 4 and 8 in. from the glass

on both sides. On the cold side there were also couples at 16 in. and 54 in. from the glass. Two thermocouples were placed in the air space with junctions against the surfaces of the two lights of the double glazed window, on the same perpendicular line with the gradient couples. In case of the wider spacings, ½ and ⅝ in., one was also placed in the center of the air space.

The same gradient couples were used for the tests with the 36-light window. They could not be located, however, on the perpendicular to the center of the window, since the center was the intersection of two muntins. The light im-

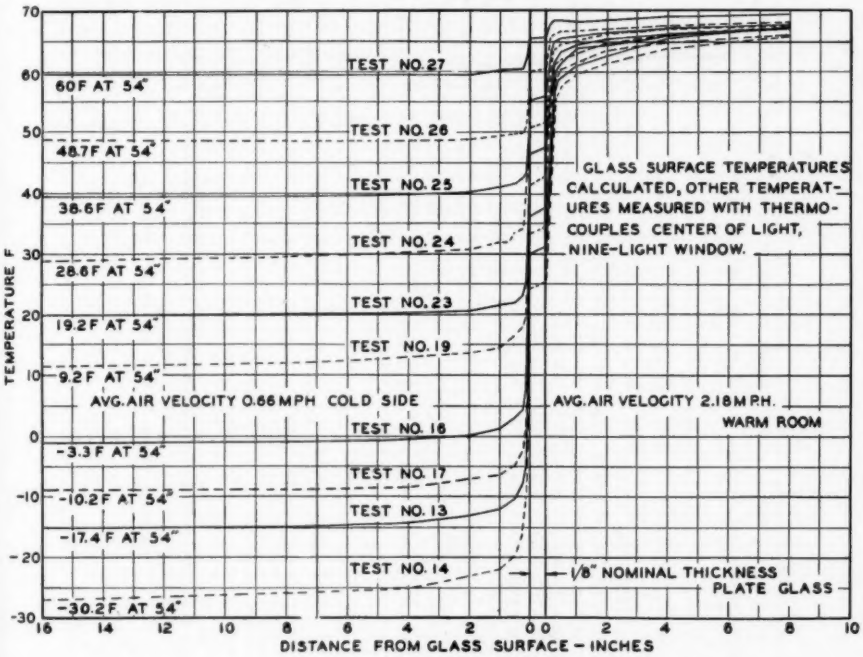

FIG. 3. TEMPERATURE GRADIENTS THROUGH ⅛-IN. PLATE GLASS SINGLE GLAZED IN THE NINE-LIGHT WINDOW

mediately above and to the right of the center was chosen, and the couples were placed on the perpendicular to the center of this light. Also, it was not possible to accommodate sufficient couples to place one opposite each light of the 36-light window. Thirteen were used, disposed as uniformly as possible.

Temperatures were measured at a number of other points in the Cold Room, the Guarded Enclosure, and the Warm Room, at least one mercury thermometer being placed in each. Following each test, all the temperature readings of each thermocouple after equilibrium had been reached were averaged and the averages used in the computations.

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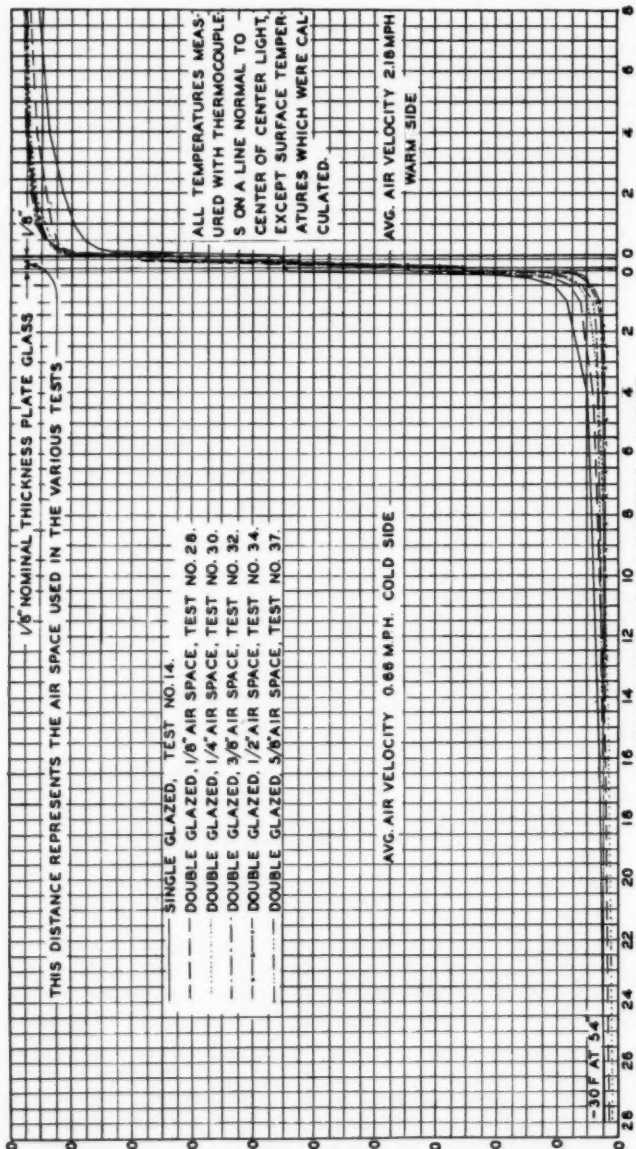
1/8" NOMINAL THICKNESS PLATE GLASS
1/8"
THIS DISTANCE REPRESENTS THE AIR SPACE USED IN THE VARIOUS TESTS
SINGLE GLAZED, TEST NO. 14.
DOUBLE GLAZED, 1/8" AIR SPACE, TEST NO. 28.
DOUBLE GLAZED, 1/4" AIR SPACE, TEST NO. 30.
DOUBLE GLAZED, 3/8" AIR SPACE, TEST NO. 32.
DOUBLE GLAZED, 1/2" AIR SPACE, TEST NO. 34.
DOUBLE GLAZED, 5/8" AIR SPACE, TEST NO. 37.
AVG. AIR VELOCITY 0.66 M.P.H. COLD SIDE
ALL TEMPERATURES MEASURED WITH THERMOCOUPLES ON A LINE NORMAL TO CENTER OF CENTER LIGHT, EXCEPT SURFACE TEMPERATURES WHICH WERE CALCULATED.
AVG. AIR VELOCITY 2.18 MPH WARM SIDE
-30 F AT 34
TEMPERATURE °F
DISTANCE FROM GLASS SURFACE - INCHES

FIG. 4. TEMPERATURE GRADIENTS THROUGH ⅛-IN. PLATE GLASS DOUBLE GLAZED WITH VARIOUS AIR SPACES IN THE NINE-LIGHT WINDOW

TEMPERATURE GRADIENTS

A few of the temperature gradients for the tests of the nine-light window are reproduced in Figs. 3, 4 and 5. These are included merely as samples of this type of data, and are characteristic of all the tests. The glass surface temperatures were computed from the air temperature in the rooms, the heat flow, and the film conductance coefficient for the prevailing mean temperature as given by the findings of Rowley [1] and the Research Laboratory.[2]

All other points were observed. The largest part of the temperature difference between air and glass in either room occurred within a distance of ¼ in. from the glass surface. The thermal resistance of the glass is small. Comparison of Figs. 3 and 5 shows that inside surface glass temperatures are sensibly higher in the double glazing tests than in the single glazing test. In the single glazed tests these temperatures ranged from 25.5 F to 65.5 F, while in the double glazed tests the range was from 46.5 F to 67.5 F.

HEAT LOSS

Heat flow data are given in Table 1 and in Figs. 6, 7 and 8. The single glazed, nine-light window was run through a complete series of tests. Tests on the nine-light, double glazed window were run at cold room temperatures of approximately −30 F and 0 F, and with the spacing varied in an endeavor to determine the best spacing which was concluded to be ¼ in. A complete set was run at this spacing. The 36-light window was tested single glazed with the cold room at approximately −20 F, 10 F and 30 F, and double glazed with ¼-in. spacing at cold room temperatures of −30 F, 5 F, and 30 F.

The transmittance coefficients for the different arrangements of sash, expressed in Btu per hour per degree temperature difference between the air in the warm and cold rooms per square foot of window opening, are given in Fig. 6 for various mean temperatures. These values are plotted against the mean temperature. The curves for the nine-light windows are rather well determined. Those for the 36-light window are less exact, as only three points were determined. These curves were fitted by the method of least distances.

At 20 F mean temperature, the nine-light window, double glazed with ¼-in. spacing, showed only 59.5 per cent of the heat loss of the same window single glazed. At 70 F mean temperature, the loss was 62.8 per cent. For the 36-light window, the figures were 62.6 per cent at 20 F, and 66.1 per cent at 70 F. Curves giving the calculated heat transfer for single and double glass based upon accepted inside and outside conductance coefficients for 2.18 mph and 0.66 mph respectively, and for accepted air space conductances based upon the work of Rowley [3] and the Research Laboratory [4] are also given in Fig. 6.

Fig. 7 shows the variation of heat loss and percentage saving as affected by the thickness of the air space. As would be expected, an increased air space decreased the heat loss. But the curve flattened so rapidly that increases in the width of the air space above ⅜ in. produced relatively small changes in

[1] Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429).

[2] Wind Velocity Gradients Near a Surface and Their Effect on Film Conductances, by F. C. Houghten and Paul McDermott. (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 301).

[3] Loc. cit. See Note 1.

[4] Loc. cit. See Note 2.

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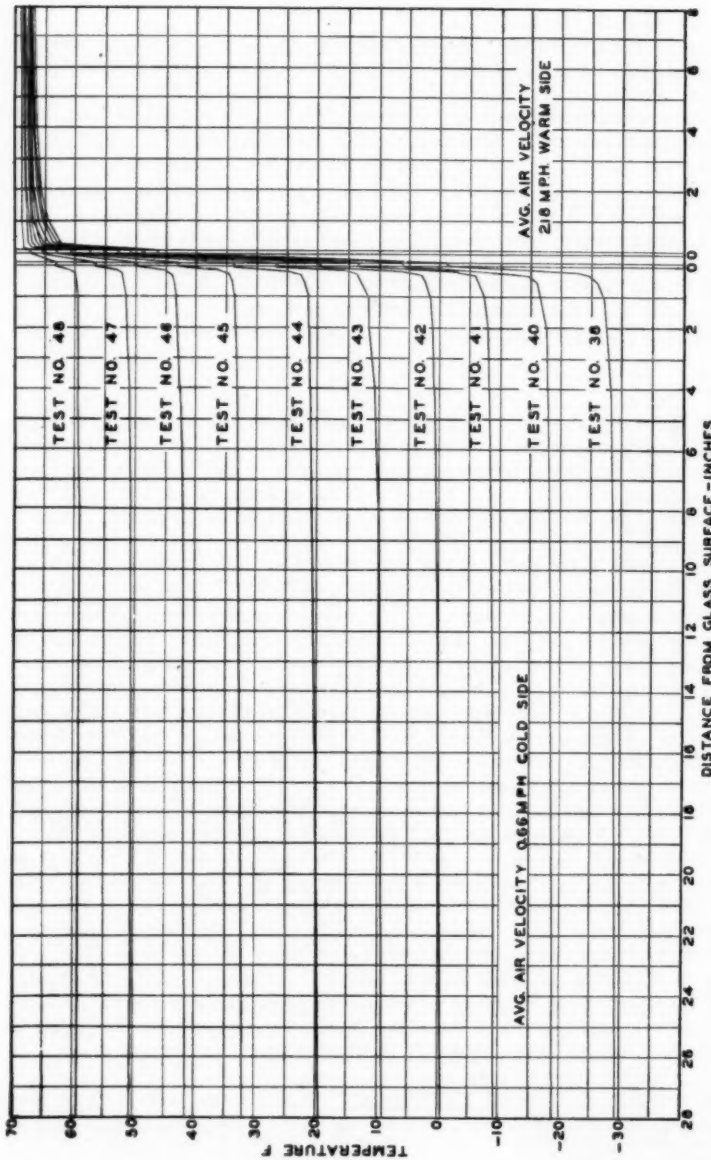

FIG. 5. TEMPERATURE GRADIENTS THROUGH 1/8-IN. PLATE GLASS DOUBLE GLAZED WITH 1/4-IN. AIR SPACE IN THE NINE-LIGHT WINDOW

the heat flow. This fact, and the dimensional limits of commercial frames, guided the choice of ¼ in. for the majority of the double glazing tests. It would seem that air spaces between ¼ and ⅜ in. would be most practical. The

TABLE I—HEAT FLOW DATA

TEST NO.	AIR SPACE IN.	TEMPERATURE, DEG F				DIFF.	ACTUAL MEAN	BTU PER HR	
		WARM ROOM		COLD ROOM				TOTAL	PER DEG F DIFF. PER SQ FT
		ACTUAL	ADJUSTED	ACTUAL	ADJUSTED				
NINE-LIGHT WINDOW SINGLE GLAZED									
14		70.2	70.0	-30.0	-30.2	100.2	20.1	5380	0.990
13		70.4	70.0	-17.0	-17.4	87.4	26.7	4651	0.981
17		70.6	70.0	- 9.6	-10.2	80.2	30.5	4269	0.981
16		72.9	70.0	- 0.4	- 3.3	73.3	36.3	3907	0.983
19		70.9	70.0	10.1	9.2	60.8	40.5	3285	0.996
23		71.3	70.0	20.5	19.2	50.8	45.9	2903	1.053
24		71.1	70.0	29.7	28.6	41.4	50.4	2182	0.974
25		70.7	70.0	38.3	38.6	31.4	55.0	1793	1.053
26		70.6	70.0	49.3	48.7	21.3	60.0	1229	1.084
27		70.7	70.0	60.7	60.0	10.0	65.7	563.5	1.040
NINE-LIGHT WINDOW DOUBLE GLAZED									
28	1/8	70.4	70.0	-29.0	-29.4	99.4	20.7	3517	0.853
29	1/8	70.5	70.0	5.2	4.7	65.3	37.8	2315	0.854
30	1/4	70.8	70.0	-30.2	-31.0	101.0	20.3	3159	0.577
31	1/4	70.2	70.0	1.1	0.9	69.1	35.7	2285	0.810
32	3/8	71.6	70.0	-31.0	-32.8	102.8	20.3	3125	0.562
33	3/8	70.8	70.0	2.3	1.5	68.5	36.5	2059	0.555
34	1/2	69.9	70.0	-30.0	-29.9	99.9	20.0	2966	0.548
35	1/2	70.2	70.0	- 0.9	- 1.1	71.1	34.6	2135	0.553
37	5/8	70.6	70.0	-30.9	-31.5	101.5	19.8	2863	0.520
36	5/8	70.2	70.0	0.1	- 0.1	70.3	35.0	2062	0.542
38	1/4	71.5	70.0	-29.0	-30.5	100.5	21.3	3177	0.583
40	1/4	70.8	70.0	-19.4	-20.2	90.2	25.7	2855	0.584
41	1/4	70.4	70.0	- 9.8	-10.2	80.2	30.3	2626	0.603
42	1/4	71.1	70.0	- 0.2	- 1.3	71.3	35.5	2315	0.599
43	1/4	70.9	70.0	9.3	8.4	61.6	40.1	1998	0.597
44	1/4	70.6	70.0	19.2	18.6	51.4	44.9	1720	0.618
45	1/4	70.8	70.0	32.2	31.4	38.6	51.5	1314	0.627
46	1/4	69.9	70.0	41.4	41.5	28.5	55.7	1005	0.651
47	1/4	69.9	70.0	50.9	51.0	19.0	60.4	6782	0.658
48	1/4	70.1	70.0	60.2	60.1	9.9	65.1	3456	0.651
THIRTY-SIX-LIGHT WINDOW SINGLE GLAZED									
52		69.5	70.0	-20.7	-20.2	90.2	24.4	5378	1.099
51		69.6	70.0	9.0	9.4	60.6	39.3	3579	1.089
53		70.0	70.0	34.5	34.5	35.5	52.2	2195	1.141
THIRTY-SIX-LIGHT WINDOW DOUBLE GLAZED									
57	1/4	70.1	70.0	-28.2	-28.3	98.3	21.0	3645	0.664
56	1/4	70.3	70.0	5.3	5.0	65.0	37.8	2475	0.697
58	1/4	70.5	70.0	33.5	33.0	37.0	52.0	1485	0.740

data for heat flow found in the single glazing tests are plotted for air space equal to zero.

EFFECTS OF AIR VELOCITY

All the tests discussed previously were made under the same air velocity conditions; *i.e.*, with the fan in the Cold Room blowing at the cooling coils from the center of the room and the two 10-in. oscillating fans in the Guarded Enclosure blowing over the heaters toward the shield. All fans were operated under the identical conditions during heat flow tests as they were when subse-

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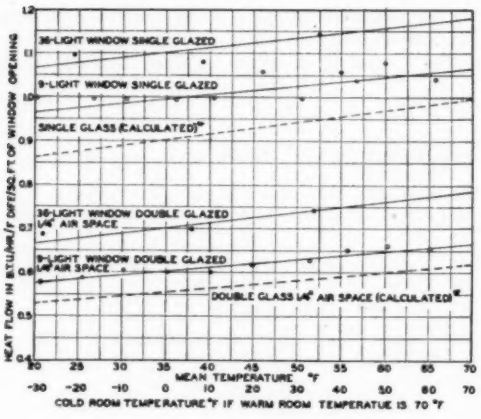
36-LIGHT WINDOW SINGLE GLAZED
9-LIGHT WINDOW SINGLE GLAZED
SINGLE GLASS (CALCULATED)*
36-LIGHT WINDOW DOUBLE GLAZED 1/4" AIR SPACE
9-LIGHT WINDOW DOUBLE GLAZED 1/4" AIR SPACE
DOUBLE GLASS 1/4" AIR SPACE (CALCULATED)*
HEAT FLOW IN BTU/HR./F DIFF/SQ.FT OF WINDOW OPENING
MEAN TEMPERATURE °F
COLD ROOM TEMPERATURE °F IF WARM ROOM TEMPERATURE IS 70 °F

FIG. 6. RELATION BETWEEN MEAN TEMPERATURE AND THERMAL TRANSMITTANCE OF SINGLE AND DOUBLE GLAZED PANELS

* Calculated curves are based on surface coefficients as given by Rowley [5] and the Research Laboratory,[6] and air space conductances given on page 104 of THE A.S.H.V.E. GUIDE 1937.

quent tests showed average air velocities 1 in. from the lights of 0.66 mph and 2.18 mph in the Cold Room and in the Warm Room, respectively. The heat flow values given in Figs. 6 and 7 are for these velocities.

Early in the course of the tests it became apparent that the velocity of the air passing over the window had a marked influence on heat loss. While not included in the original plan of the investigation, the importance of the subject led to the decision to do some preliminary work with different air velocities. Accordingly, a few temperature measurement tests and heat loss tests were

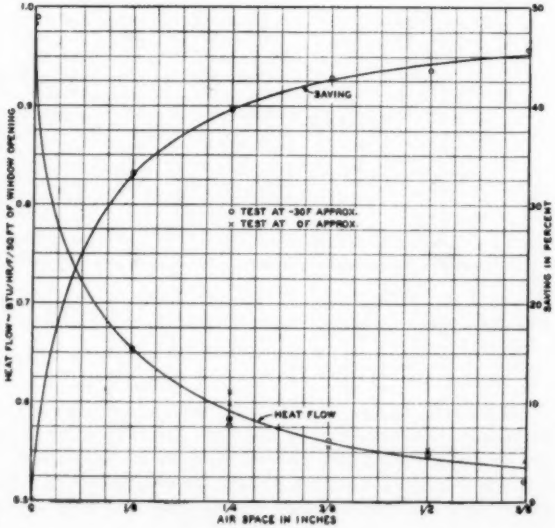

FIG. 7. THERMAL TRANSMITTANCE AND HEAT FLOW SAVING AS FUNCTIONS OF AIR SPACE —NINE-LIGHT WINDOW

[5] Loc. cit. See Note 1.
[6] Loc. cit. See Note 2.

made following the procedure previously described but with different air velocities and of shorter duration. The velocities used in the Cold Room were zero, 0.66 mph and 7.58 mph; in the Guarded Enclosure the velocities were zero and 2.18 mph. These are the average velocities measured 1 in. from the

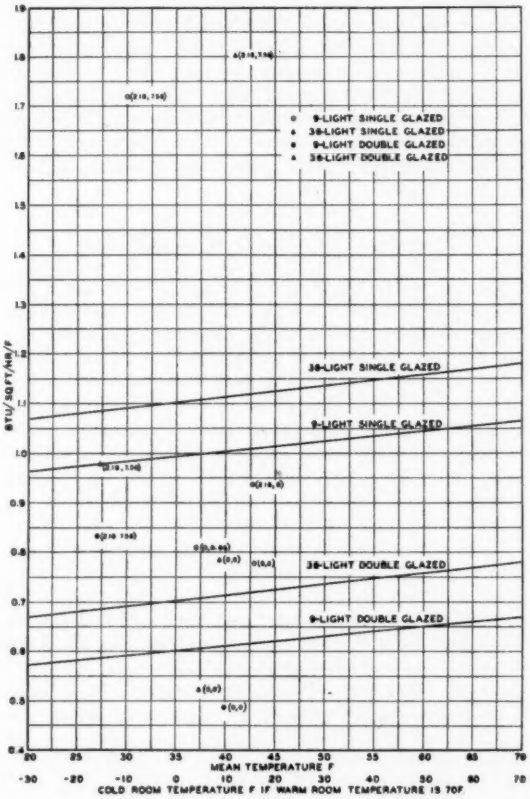

FIG. 8. EFFECT OF AIR VELOCITY ON THE THERMAL TRANSMITTANCE OF THE WINDOWS

glass with a Kata thermometer. Zero velocity was had with no fans running; actually, there probably was a motion due to convection.

The results of the velocity tests as far as temperature effects are concerned were reasonable and as would be predicted. Air temperature gradients in the Cold Room were flatter with higher air velocities and had a more abrupt transition at the surface film. The distribution of temperatures over the window, measured 1 in. away from the center of each light, was more uniform. In the

Guarded Enclosure, the gradients showed no great difference between *still air* and the rather low velocity of 2.18 mph.

The data on heat loss as affected by air velocity are given in Fig. 8. The curves are reproduced from Fig. 6. In all tests represented by the curves, the Cold Room average velocity was 0.66 mph, while the average velocity in the Guarded Enclosure was 2.18 mph. The points for other velocities plotted on the chart represent the heat transfer for the particular window indicated by the symbol and for the air velocities given in the associated parentheses. The velocity in the Guarded Enclosure is given first. The vertical separation between any point and the curve for the same window similarly glazed represents the change in heat flow attributable to the change in air velocities. This does not represent an exhaustive study of the subject, and the data are too scant to permit of any quantitative conclusions. They do indicate the necessity for further work on the subject.

MOISTURE AND FROST FORMATION

During the tests of the single glazed window considerable difficulty was experienced in keeping frost from forming on the warm side of the window when the Cold Room temperature was −30 F. Silica gel was used to absorb moisture in the Guarded Enclosure, but it was found that the humidity could not be reduced below 30 per cent. Whenever the temperature of the warm surface of the glass fell below the dew point of the air in the Guarded Enclosure, moisture formed on the glass. This abstraction of moisture from the air lowered the dew point so that only small amounts of dew or frost formed, provided no fresh air was admitted to the Guarded Enclosure by the opening of the door.

Notes made during the single glazed tests show that with a Cold Room temperature of −30 F thin frost was distributed at random over the entire window, the total area covered being about 1 sq ft. At a Cold Room temperature of −20 F, almost all the frost had disappeared. When the Cold Room was at −12 F, and when there was an average velocity of 7.58 mph on the cold side, a very thin film of frost formed on the warm surface over about one-fourth the area of the window. No condensation was noted on the frame. It should be remembered, however, that these observations were made with humidity conditions on the warm side as low as possible, a condition which seldom prevails in buildings.

No frost formed on the warm side of the glass during any of the tests of double glazing, although no precautions were taken to keep the humidity of the Guarded Enclosure low.

Fig. 9 shows the conditions under which moisture or frost will form on single and double glazed windows. The air velocities given are, as previously, in miles per hour, with the values for the Guarded Enclosure given first. These curves do not represent experimentally observed data. They have been derived from the known properties of moist air and are based on the surface temperatures of the tests. These surface temperatures were not measured but computed from the experimentally observed heat flow and other test data.

Some observations were made concerning condensation in the air space of the double glazed windows. At Cold Room temperatures of −30 F, −20 F, −10 F, and −1 F, frost formed on the cold glass surface in the air space

of every light. At 9 F the frost was melting, moisture remaining as a thin film. At 19 F all the frost had melted, running drops had formed. At 32 F the moisture occurred as a thin film with a few drops. At 42 F some moisture was still noticeable as a thin film. At 52 F, it was practically all gone, and at 60 F, the window was clear. Twenty-four hours elapsed between the successive temperature changes listed above.

Venting of the air space was accomplished by removing a 1-in. section from the bottom aluminum spacer of each light, the putty being removed to vent the air space to the cold side. One five-hour test at −30 F was run with the vents open. Based on the results of this one test, the heat loss increased between 2 and 3 per cent. The results with respect to the removal of frost from the cold glass surface inside the air space were not as good as expected. Twenty-four hours after the vents had been opened with the Cold Room remaining at

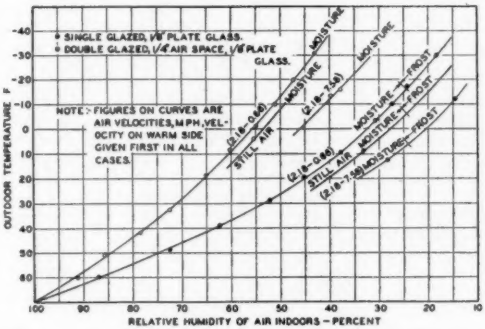

FIG. 9. CURVES OF MOISTURE FORMATION AT VARIOUS AIR VELOCITIES

−30 F, almost, but not all, of the frost had disappeared from the upper lights, but almost all of it still remained on the bottom row of lights. It was determined later that if the vents were opened before the frost formed results were better. It should also be noted that after resealing the vents almost as much frost formed as previous to opening them, although considerable time was required. The subject of venting should receive further investigation.

Transmittance coefficients for single and double glazed windows resulting from investigations by various authorities vary considerably. Values used in THE A.S.H.V.E. GUIDE since 1922 show variations ranging from 1.03 to 1.13 for single glazed or single sash windows, and from 0.60 to 0.45 for double glazed or double sash windows; the last value in each case being used in THE GUIDE since 1929. John R. Allen [7] gives a value of 0.64; Arthur N. Sheldon [8] gives values of 1.32, 0.88, 0.98, 1.13 and 0.61 for single glazed steel, double glazed steel with ¾-in. air space, double glazed steel with 3/16-in. air space, single glazed wood, and double glazed wood with ⅝-in. air space, respectively.

[7] Heat Transmission Through Building Materials, by John R. Allen. (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916, p. 515).

[8] Heat Transmission Through Various Types of Sash, by Arthur N. Sheldon. (*A.S.M.E. Transactions*, Vol. 38, 1916, p. 869.)

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Hammond and Holmberg [9] give transmittance coefficients ranging from 0.97 to 1.16 for variations in mean temperatures from 52 to 86 F. A calculation based upon the film conductance coefficients resulting from work of Rowley [10] and verified by the work of the Research Laboratory,[11] for still air with the prevailing mean temperature and with zero degrees outside and 70 F inside, and accepting what is recognized as a high value for the conductivity of glass, gives 0.688 for a ⅛-in. thickness of glass. The relatively small effect which the resistance of the glass itself offers is indicated by the fact that in completely neglecting the effect of the glass in this computation the coefficient of 0.698 is indicated. Applying a similar computation and using the air space coeffi-

TABLE 2—COMPARISON OF RESULTS WITH THOSE DERIVED FROM CALCULATION

No.	Description	Air Velocity (mph)		Mean Temp. F	Transmittance Btu/Hour/F/Sq Ft	Source of Data
		Warm Side	Cold Side			
1	36–light single glazed...	2.18	0.66	35	1.10	This Paper
2	Nine–light single glazed	2.18	0.66	35	0.99	This Paper
3	Glass only.....	2.18	0.66	35	0.90	Calculated[a]
4	Glass only.....	Still air	15	35	1.14	Calculated[a]
5	Glass only.....	Still air	15	35	1.13	A.S.H.V.E. Guide
6	36–light double glazed..	2.18	0.66	35	0.71	This Paper
7	Nine–light double glazed	2.18	0.66	35	0.60	This Paper
8	Glass only.....	2.18	0.66	35	0.56	Calculated[a]
9	Glass only.....	Still air	15	35	0.63	Calculated[a]
10	Glass only.....	Still air	15	35	0.45	A.S.H.V.E. Guide

[a] Calculation based on a glass conductivity of 7.2 Btu per hour per inch per degree Fahrenheit per square foot and the film conductance coefficients and air space conductances given by Rowley and the A.S.H.V.E. Research Laboratory as referred to in footnotes 1 and 2.

cients given in THE GUIDE, two ⅛-in. thicknesses of glass with a ¼-in. air space, gives a coefficient of 0.46.

The considerable variation in these values from different authorities may be accounted for by possible errors in some of the earlier work due to correction for box losses and due to variations in the air velocities actually existing near the glass surface. The calculated values based upon film conductance coefficients are for no air velocity whatever. The values reported by Allen are for velocities existing with no artificial air movement in and around his testing box, while the values reported by Hammond and Holmberg and by Sheldon are for conditions where an unreported artificial air motion was supplied within the box. The values now used in THE GUIDE apply for natural convection within the room and a 15 mph wind outside.

The transmittance coefficients for the several window arrangements studied and for several conditions calculated from film conductance coefficients and air

[9] A Study of Surface Resistance with Glass as the Transmission Medium by H. R. Hammond and C. W. Holmberg. (*Journal* of the *A.S.M.E.*, Oct. 1917, p. 849.)
[10] Loc. cit. See Note 1.
[11] Loc. cit. See Note 2.

space conductances are given for comparison in Table 2. Comparisons of conditions 1 and 2 with 3 for single glazing and 6 and 7 with 8 for double glazing show that calculated values based upon accepted surface coefficients and the same wind velocities on the inside and outside give lower values for glass than found for the panels tested in this study. This would tend to show that, under the conditions of test, the size of the individual lights in the test panel bears a definite relation to the heat transfer, and that the greater the size of the individual lights, the less the loss per square foot.

CONCLUSIONS

1. Double glazing reduces heat losses materially as compared with the losses which would take place through the same windows single glazed, under the same conditions of temperature and air movement. The tests made in which only low air velocities were used showed reductions in heat flow ranging from 33 to 46 per cent, depending on the thickness of the air space.
2. Increasing the spacing between the lights up to ⅜ in. showed a rapid decrease in the rate of heat transfer. A further increase in the spacing produced relatively small change in the heat flow. This fact, together with the dimensional limits of commercial frames, indicates that spacing from ¼ in. to ⅜ in. is the optimum spacing for double glazing. The tests showed about a 40 per cent reduction in heat loss with ¼ in. spacing, as compared with the heat transfer for the same window single glazed and under the same conditions of temperature and air movement.
3. The few tests made to show the effect of air velocity indicated a very marked effect in increasing heat transfer through windows. This increase was much greater for the single than for the double glazed windows.
4. Double glazing results in higher indoor glass surface temperatures, as indicated by the absence of moisture or frost formation, thus resulting in greater comfort for the occupants of the room.

DISCUSSION

P. F. MCDERMOTT[12] (WRITTEN): The study reported is well chosen and has been well done. As noted in the paper further work is needed to broaden the applications of the results. Several suggestions may be of aid in doing further work along these lines:

1. The use of aluminum spacers between the two plates of glass in the double-glazed windows seems undesirable because of the high thermal conductivity of the aluminum. In terms of the cross-sectional area of the metal, as a path for heat flow, when compared with the total area of the glass, little effect might be anticipated. However, the spacer being in good thermal contact with both glass plates which themselves are good conductors, considerable heat can be carried from the hot to the cold glass without traversing the air space at all and thereby materially decrease the efficiency of the double glazing. Wooden spacers appear to be more desirable especially if made of maple or mahogany well sized with oil, in which case accurate dimensions with a minimum of variation with moisture content would be had.
2. The authors make no mention of what conditions obtained when air velocities were determined by the Kata thermometer. It is probable that consideration was given to the temperature of the glass in correcting for the temperature of the surroundings during Kata measurements. A further precaution to prevent error would be to compare the results obtained by both wet and dry instruments. The

[12] Physicist, Johns-Manville Research Laboratories, Manville, N. J.

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error introduced, if no precautions were taken, would result in higher indicated velocities on the warm side and lower ones on the cold side of the window.

Observations in connection with condensation of moisture were of interest. Considerable damage to window frames and sash frequently results from this cause. A method for properly applying storm sash in such a way as to prevent condensation and at the same time conserve heat is desirable.

The prevention of condensation by venting to the outside is well established in the construction of refrigerated equipment and more recently is gaining recognition in building construction. The inability to remove frost, once formed, by venting has been observed. The vent application must be a permanent one which will prevent the formation of liquid or solid condensate. The slight increase in the heat flow produced by the large vents used in the study demonstrates how little air need be circulated and therefore how little the heating load is increased.

Data should be gathered in a similar manner on standard windows as used in dwellings and should include standard weatherstripping and storm sash treatments, etc. Such data are more and more important in figuring heating loads as the walls of the structure are better insulated. For example calculating heat losses for a frame structure insulated with 4 in. of rock wool shows in the neighborhood of 60 to 80 per cent of the loss to occur through windows, doors, and by infiltration. Frequently the heating system is thus selected to fit the windows and doors. Hence, if data concerning them are not accurate, heating designs will be subject to too frequent failure.

PROF. C. M. HUMPHREYS (WRITTEN): The authors of this paper are to be congratulated for discovering what appears to be a serious error in our present heat transmission coefficients when applied to some common types of windows. Although the paper offers very little toward the final solution of the problem, it does indicate its nature. This should assist greatly in planning a research program which will lead to the correct solution.

This paper mentions three factors which influence the heat transmission through windows; namely, the size of lights, the wind velocity, and in double glazing, the width of the air space. These three factors immediately suggest others. If a research program is planned to make a complete investigation of the flow of heat through windows, the following studies are sure to be included: effect of wind velocity; effect of muntin spacing, both horizontal and vertical; effect of muntin shape; effect of material used for muntins; and effect of depth of reveal.

After the effect of each of the various factors is thoroughly understood, there will still remain the problem of expressing the combined effect of these factors in a way that will be reasonably accurate and yet readily usable by the designing engineer. Unless this final goal is achieved, any effort spent in the study of the problem will be wasted.

Most of the tests described in this paper were made with air velocities on the warm and cold sides of the windows of 2.18 and 0.66 mph respectively. A few tests were made at other velocities, and the points determined by these tests are shown on Fig. 8 of the paper.

To get a clearer picture of the effect of wind velocity on heat transfer through glass, some of the points shown on Fig. 8 have been replotted using wind velocity on the cold side as abscissa and Btu per square foot per hour per degree Fahrenheit as ordinate. The value of each of these points was corrected to 35 F mean temperature by drawing a line through the point parallel to the curves for the corresponding window, and reading the value at the intersection of this line and the 35 F mean temperature line.

These points, plotted to the new coordinates, roughly locate four curves, one each for nine-light and 36-light single-glazed windows, and one each for nine-light and 36-light double-glazed windows, as shown in Fig. A. It is emphasized that these curves cannot be considered accurate, because of insufficient data. The curves are

plotted with only two points each, and two points do not determine a curve. If the curves do no more than direct attention to the importance of air velocity in heat transfer through windows, they will have served their purpose.

Four calculated curves are also shown on Fig. A for the purpose of comparison. These show the calculated values of heat flow through single and double glazing for air velocities on the warm side of the glass of zero and 2.18 mph. The surface and air space conductances used in calculating these values were obtained from the same

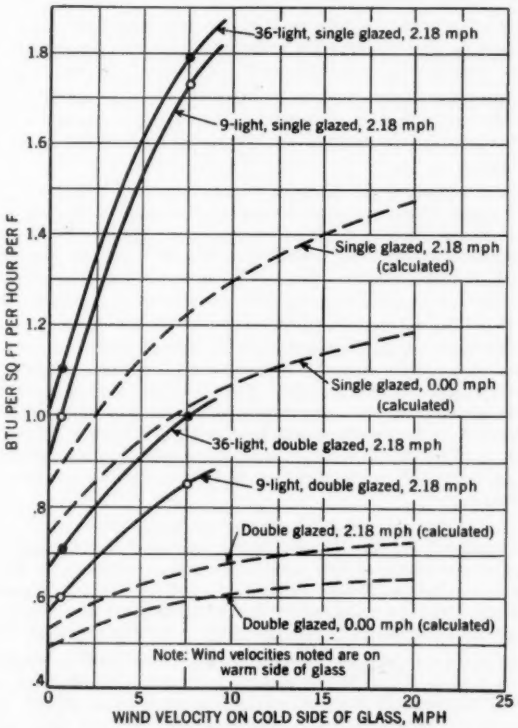

FIG. A. EFFECT OF AIR VELOCITY ON THE THERMAL TRANSMITTANCE OF WINDOWS

sources used in the original paper for similar values. The large difference between the observed and the calculated values are evident.

There is still another phase of the air velocity problem about which more information is needed. In addition to knowing the effect of a certain air velocity across the glass, we should also know what air velocity is likely to occur. In all studies of the effect of air velocity on surface conductances, the velocities have been measured within an inch or two of the surface. Then for no reason whatsoever, we assume that the average wind velocity for the heating season as measured by the U. S. Weather Bureau on top of some building will also be the air velocity across the glass.

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A paper [13] on this particular subject seems to indicate that the actual wind velocity near a building is much lower than the velocity as measured at the Weather Bureau. Further research is needed on this subject.

T. S. ROGERS [14] (WRITTEN): As a lay member of the Technical Advisory Committee on Insulation, it is my function to examine developments in insulation research from the point of view of the public rather than in terms of technical procedures. The public I construe to include architects, contractors, building owners and others who put into practice the findings of research authorities.

From this point of view the present study of heat transfer through glass must be looked upon as a definite contribution toward much needed knowledge. At the same time it has limitations which require careful handling of the resulting data.

Perhaps of greatest significance is the fact that this research in double glazing further confirms the impression of many practical laymen that the heat transfer data in THE A.S.H.V.E. GUIDE are often too theoretical and do not always reflect actual service conditions.

Practically all heat transfer information now published is based upon the performance of samples of materials which are studied without respect to their manner of assembly in actual buildings. It is of utterly no moment to a heating engineer if the transmission coefficient for single glass is 1.13 when the glass is disassociated from a window, if that coefficient does not apply when the same glass is mounted in a wood or metal sash as part of a complete window area. Heating contractors must calculate losses through windows, not through glass alone.

This research indicates that transmittance is related to the size of the individual lights. It further indicates that the value of 1.13 for single glass is probably safe when the glass is installed in steel sash. On the other hand it shows that the value 0.45 for double glazing may be substantially lower than is attained when the double glazing is mounted in steel sash of standard section. In short, the present research emphasizes the need for much more comprehensive study on the subject in order that values used in computations may reflect service conditions ordinarily encountered in building practice.

Obviously the performance of glass in windows where the sash is of wood is likely to vary from windows of similar area when the sash is of metal because of the different conductance of metal and wood. More than half of our heat conservation problem is encountered in residences and small buildings. We need much more information about heat losses through the ordinary residential double-hung wood sash, wood casement, and metal casement than is to be found in data so far published.

The present report has its own limitations as a contribution in this direction. For some reason not made clear in the paper, the test conditions were the reverse of those usually encountered in the use of double glazing in windows. In this test the average air velocity on the cold side was negligible whereas that on the cold side is generally assumed to reach a velocity of 15 mph. The test figures given in the paper might well apply to refrigerated show cases in stores where there is practically no air movement within the refrigerator case and a moderate movement in the warm air surrounding it. But it would hardly be desirable to assume that the transmittance data published in this paper can be applied to the problem of double glazing of exterior windows.

Similarly the findings with respect to moisture and frost formation contribute little to guide us in ordinary building practices. The authors recognize the fact that the prevailing humidity conditions were quite unlike those encountered in ordinary practice.

[13] Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw, and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 387.)
[14] Glen Ridge, N. J.

It happens that the glass industry is now paying a great deal of attention to the possibility of reducing heat losses from buildings in winter by means of multiple glazing. They are encouraged in this activity by the figures long published in THE GUIDE which indicate that better than half of the heat loss through glass can be prevented by double glazing and by the data given by Prof. G. L. Larson in his addresses on The Economics of Good Construction. Professor Larson, in a study of the heat losses from his own residence, showed that the use of storm sash and storm doors without other insulation gave a higher net return on the investment than any other means he studied for checking heat losses.

I think the industry has every justification for seeking to educate the public in the advantages of multiple glazing of windows. But I am also conscious of the fact that we seem to have inadequate information as to the allowances which can be made in the size of the heating plant or in the fuel savings to be anticipated when double glazing is used. The present paper is noteworthy as being one of the very few reports contributing to the scant information now in existence. Its findings as well as its limitations should encourage much further work in this field.

PROF. E. R. QUEER (WRITTEN): There are two important points raised in this paper which should be emphasized. First, the matter of condensation and frost on windows is quite troublesome, particularly in double glazing. Second, the heat transfer through double-glazed windows is greater for small-size panes than large-size panes.

The tendency at present is to artificially introduce moisture into residences during the heating season, thus improving the comfort of the occupants. Among other things, this gives rise to condensation and frost difficulties on the windows. In the case of double glazing instances are known where condensation has fogged the windows early in the heating season and the condition persisted until spring. Venting the space between the panes to the outdoors relieved the condition somewhat. This subject should receive further investigation as suggested by the authors.

There are two factors which account for the heat transfer through double glazing being larger for small-size panes than for large panes. First, it has been found that breaking up an air space into small units (reducing the height) increases the heat flow.[15, 16] A 20-in. height of air space will be more efficient in this respect than a 15-in. height. Beyond a 20-in. height the heat transfer will decrease slightly up to 35 in., where it becomes almost constant. Second, small panes require more muntin area in a given window area. If metal muntins, having a relatively high thermal conductivity, are used, they will tend to increase the heat transfer value. The metal forms a short-circuit heat path through the window.

There are several minor questions of academic interest which might be raised regarding the apparatus and operation of the equipment. First, how much of a temperature variation was observed within the test box, the guard box, and the cold room during tests at high heat inputs? Second, it is noted that the thermocouples were placed in direct contact with the glass; but the temperatures obtained from these couples were not used as the surface temperature of the glass. Did these values deviate much from the calculated surface temperatures? Third, were the refrigeration coils in the cold room screened from the specimen?

PROF. F. B. ROWLEY (WRITTEN): The results as presented by the authors in this paper are of interest for several reasons. They indicate the care which must be used when applying fundamental constants to practical building construction. An analysis of the test results indicates that in most cases they are not materially different than would be expected from the correct application of fundamental constants previously

[15] Importance of Radiation in Heat Transfer Through Air Spaces, by E. R. Queer. (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 77.)

[16] Notes on the Testing of Aluminum Foil Insulation, by J. D. Babbitt. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1937.)

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determined, and the conclusions reached agree with the conclusions from previous investigations.

A comparison of the heat transfer coefficients as obtained by test and by calculation, and shown in Fig. 6, might at first thought, indicate wide variation in the fundamental constants which have been determined for surface and air space coefficients and those which may be expected from actual construction. The test values are much higher for both the single- and double-glazed sash than are the calculated values.

The over-all resistance to heat flow through a single-glass window is practically all confined to the surfaces. Any irregularities or projections on these surfaces will increase their heat transmitting area per unit area of window and therefore increase the heat flow. The surface areas of the mullion bars extended on each side of the glass for those windows tested add materially to the total surface area, and since these projections are connected with high conductivity material the over-all heat transmission coefficient should be increased almost in proportion to the increase in surface area caused by the mullion bar extensions. The area of the mullion bar extensions is substantially twice as great in the 36-light window as for the nine-light window. Therefore the increase in the coefficient for the 36-light window should be substantially twice as much as for the nine-light window. The effect of the mullion bar extensions will not be as great for the double-glazed windows, since in these cases the outer and inner surface resistances do not form as large a percentage of the total or over-all resistance as they do for the single-glass windows.

The test values in the curves of Fig. 6 were determined for air velocities of 0.66 mph on the cold side and 2.18 mph on the warm side of the glass. In order to compare them with calculated values a point at 40 deg mean temperature will be taken and coefficients will be selected from previous test data[17, 18] at the corresponding mean temperature and wind velocities. The coefficients selected from the given references are:

f_0 = 1.5 for glass at 20 F mean temperature and 0.66 mph wind velocity.[17]

f_1 = 2.1 for glass at 60 F mean temperature and 2.18 mph wind velocity.[17]

a = 1.47 for a ¼ in. air space at 40 F mean temperature.[18]

k = 8.0 for glass.
Thickness of glass = 0.125 in.

With the coefficient stated the over-all calculated conductivity values are as follows:

For single plain glass, $U = 0.863$.

For double plain glass, ¼ in. air space, $U = 0.54$.

These values are slightly lower than the corresponding calculated values given in Fig. 6, due probably to the fact that the surface coefficients were selected for different mean temperatures. Since the area of the projected mullion bars is not given, the effective area of these projections for both the nine- and 36-light windows may be calculated by assuming that all of the heat resistance for the single-glass windows is confined to the surfaces and comparing the calculated over-all coefficients to the test over-all coefficient for a particular mean temperature. The over-all coefficients will be directly proportional to the effective area in each case. The effective area

[17] Surface Conductance as Affected by Air Velocity, Temperature, and Character of Surface, by F. B. Rowley, A. B. Algren, and J. L. Blackshaw. (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429.)

[18] Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren. (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 165.)

for the calculated coefficient is unity. The effective area for the nine-light window per square foot of projected area is:

$$\frac{1.01}{0.863} = 1.16.$$

The effective area for the 36-light window is likewise,

$$\frac{1.13}{0.863} = 1.31.$$

Thus the extended surfaces of the mullion bars add 16 per cent to the effective surface areas for a nine-light window, and 31-per cent to the effective surface area for the 36-light window.

If the effective areas due to the mullion bars as calculated for the single-light windows are correct they may be used to calculate the over-all coefficient for both the nine-light and 36-light double-glazed windows and these calculated coefficients may be compared with the test coefficient.

Calculations may be made on the basis of parallel heat flow through two paths. First, through that portion of the window covered by the double glass, and second, through that portion of the window covered by the mullion bars. Since the mullion bars are of metal and therefore of high conductivity, and since the total area on each side of the glass is substantially the same, it is reasonable to assume that the heat which is conducted into the surface of the mullion bar on the hot side is carried directly through and conducted out of the surface on the cold side, and that this forms a direct path for heat flow. Likewise the heat conducted into the warm surface of the glass will flow through the double glass and out on the cold surface. On account of the area taken up by the mullion bars the net area of double glass per square foot of window area is somewhat reduced and will be taken as 95 per cent for the nine-glass window and 90 per cent for the 36-glass window. The heat flow per square foot of window area for the nine-glass window will therefore be made up of 0.95 sq ft of double glass and 0.16 sq ft of projected mullion bar area, and the heat flow per square foot of window area for the 36-light window will be made up of 0.9 sq ft of double-glass window and 0.31 sq ft of projected mullion bar area. The overall coefficient of conductance, U, for the double glass as previously calculated is 0.54, and since the projected mullion bar areas on the two sides of the glass are connected by a material of high conductivity, the overall coefficient between the surfaces will be substantially the same as that for a single-glass window under the same conditions. The calculated over-all coefficient for the nine- and 36-light windows will be as follows:

$$U \text{ (nine-light window)} = (0.54 \times 0.95) + (0.863 \times 0.16) = 0.651$$

$$U \text{ (36-light window)} = (0.54 \times 0.90) + (0.863 \times 0.31) = 0.743.$$

The test coefficients from Fig. 6 at 40 F mean temperature are 0.62 and 0.73 for the nine- and 36-light windows respectively. This is an exceptionally close check and indicates that the surface coefficient as previously determined may be applied with reasonable accuracy to either single- or double-glass, metal sash windows, providing the proper surface areas are used and the construction is properly analyzed. Over-all test values are, however, assuring and necessary for many types of construction. It is impossible to take into full consideration the effect of the window frame and casing.

The conclusions drawn by the authors are amply verified by other test data, and it seems evident that calculated coefficients would agree reasonably well with the test

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coefficients if all factors could be taken into consideration when making the calculations. In continuing the work it would seem worth while for the authors to take into consideration the areas of mullion bars and see to what extent these areas would reconcile any discrepancy between test and calculated coefficients.

R. J. TENKONOHY (WRITTEN): The objectives set out by the authors are most timely, and well chosen. They might have stressed the objectives much more strongly without exaggerating the urgency for conclusive data. The conclusions, however, reveal substantial savings in double glazing with air spaces permitting the use of commercial steel frames, which results in an economy of no mean value for the building owner in cooling as well as in heating. The value of double glazing of steel frames with high humidities is also a result with far-reaching possibilities.

The authors point out that for the same wall opening a marked difference in heat loss occurs if a panel of nine lights or one of 36 lights is used. They suggest that a tendency prevails toward a definite relation between the sizes of individual lights. The magnitude of the difference indicates that some investigation is warranted. From Table 2 it appears that at the air velocities of the tests a 36-light panel, single glazed, transmits 11 per cent more heat than does a nine-light panel, and that a 36-light panel, double glazed, transmits 18 per cent more than a nine-light panel similarly glazed. This difference becomes significant in heating, and more so in cooling, when coupled with the increased transmittance at higher mean temperatures.

The authors have not given values for unit transmittance for still air inside and 15 mph outside, so that their values may be compared with those generally used in the computation of heat losses. An approximation may be made from Table 2. It might be assumed that the unit transmittance at still air and 15 mph in Lines Nos. 1 and 2 would bear the same ratio to that in Line No. 3 that the transmittance in Line No. 3 bears to that in Line No. 4 and likewise for double glazing in Lines Nos. 6, 7, 8, and 9. This leads to a unit transmittance for a 36-light panel and a nine-light panel, single glazed, with still air and 15 mph as probably 1.40 and 1.26 respectively. Similarly, double-glazed panels would probably be around 0.785 and 0.675 respectively. While these transmittance values are crudely obtained, it is not unlikely that correct values are of such order rather than 1.13 and 0.45 which are frequently used for the wall opening or panel dimension. Would the authors subscribe to such values, or have they calculated the transmittance through the panels at still air and 15 mph?

It is possible that sufficient data have been collected in the paper to show a trend in the relationship between sizes of lights and difference in heat loss through the panels. From Table 2 can be calculated the total heat per hour through glass, only, and for the entire panels. The difference would represent the probable loss through the steel frame, the muntins, and the putty. This difference, compared with the total area of panel less the glass area, would result in a figure for loss per square foot. Such procedure, applied to the data on single-glazed frame Lines Nos. 1 and 3 indicates for the 36-light panel a loss of 109 Btu per hour for an area of 0.82 sq ft, or 133 Btu per square foot. Similarly, Lines Nos. 1 and 2 for the nine-light panel indicate a loss of 65 Btu per hour for an area of 0.5 sq ft, or 130 Btu per square foot—an agreement that might show a trend.

Is it not, therefore, likely that the heat loss or heat gain of a steel framed window panel can be determined accurately from the actual exposed glass area with its unit transmittance and the area of wall opening less the exposed glass area with its unit transmittance, with both values of transmittance selected for air velocities and mean temperatures?

W. A. DANIELSON (WRITTEN): The importance of air motion is brought out strikingly in this paper. Everything points to the fact that this must not, as heretofore, be reduced to the point where it is neglected in the research. In practice it is

not absent, and the research is for the purpose of helping practice. That air motion complicates the problem is no reason for failing to get some definite information on its effect. Take away air motion from a radiator or a unit heater and the heating effect would be little more than the radiant energy emitted. Even so-called zero air motion on a vertical surface is not the case. In Fig. 8 quite a film of still air is indicated on the cold side where the velocity is slightly less than 60 fpm, and on the warm side it is negligible where the velocity is about 200 fpm. Between the panes, where there is very little movement, if any, there is a great difference between the two surfaces, but after all, this is only about three times that shown for the film of air touching the glass on the cold side and which is about the same thickness as the air between the glass. It is hoped that this Committee can carry on further studies with higher velocities at different temperatures, and especially on the cold side.

Under these test conditions it is interesting to note that windows with large panes pass less heat than those with small ones. This can be due only to the condition of the spacing bars and mullion over that of glass alone. Whatever can be done to reduce such conditions will therefore help. Of course, if the air velocity increases, then mullions, like the fins on the air-cooled engine, will more rapidly absorb heat on the hot side and give it off on the cold side. In the engine the material of which this is made is an important factor. In the engine of one manufacturer copper fins were used at one time because this metal would transmit the heat more rapidly than iron. It is hoped this Committee will get data with different materials.

The holes in the glass to overcome the condensation could hardly be expected to be effective considering the condition of the test. To get the breathing necessary to remove the moisture would require that the temperature be raised and lowered a number of times and thus simulate the changing temperature conditions in practice. With the air entering from the cold side and then warmed to lower its humidity, it will, of course, take up some of the condensed moistures and thus will be carried out when the outside air warms up. Continued repetition will gradually remove all the moisture. Had the test been started with breathing from the cold side, other data indicate there would not have been condensation. More work on this seems advisable.

The writer was, and is, strongly impressed with the possibility of taking advantage of research work done at places other than the Society's Laboratory or cooperative agencies. A thorough review of the procedure and data of such research by the Committee will undoubtedly eliminate any commercial bias that may exist. It would be ideal if the Committee's representative, who would be in the nature of an inspector on building construction, could be present while the work was under way. Funds no doubt will be available when the more important work gets started for such personnel. This use of research information available outside of the Society will speed up the needed data at a time when development is so rapid. Any minor disadvantages hardly warrant the elimination of this great fund of knowledge that is open to the Society's research activities.

PROF. T. F. ROCKWELL (WRITTEN): Presumably data resulting from investigations sponsored by this committee are intended more for the benefit of the window manufacturers rather than for the designer of the heating system. Hence the A.S.H.V.E. should use caution in giving publicity to the data contained in this report.

In cases of new construction, the designer of the heating system is unfortunately in a position where he cannot take advantage of any difference in coefficients of heat transfer which may result from the use of a different number of lights per sash, particularly when that difference is of the order of 10 per cent. The designer must of necessity use the high value. Further, Fig. 8 of the paper indicates that the difference in heat transfer between the nine-light and 36-light sash becomes less significant as outside air velocities are increased.

Fig. 7 and conclusion 2 in the paper refer to the reduction in heat loss resulting from the use of double glazing. Obviously the authors refer to the reduction for

the window area alone, but they do not clearly say so. Uninformed persons might jump to the conclusion that adopting double glazing would enable them to reduce their fuel consumption by 40 per cent when such is actually not the case. Heat loss through glass usually acounts for 30 per cent to 40 per cent of the total heating load in buildings which are not ventilated. Saving 40 per cent of this fraction would only permit a theoretical reduction in the fuel consumption of 12 per cent to 16 per cent.

In school houses where the heat required to temper outside air used for ventilation may account for over one-half of the total demand, the glass loss will be less than one-sixth of this total. Double glazing would produce a still smaller per cent saving in this type of building.

The foregoing is, however, not to be construed as an argument against double glazing. Figs. 3 and 5 indicate that with double glazing inside glass surface temperatures are from 17 to 20 F higher than for single glazing. Not enough architects and others responsible for building construction realize the connection between these surface temperatures and bodily comfort. Maintaining the radiant heat loss from the body at its proper level does away with the desire for overheating air above 70 F. There is, of course, a potential fuel saving if overheating can be eliminated, but its magnitude is difficult to forecast, and in many cases the control system should be credited with part of this saving.

Double glazing also, because of this higher inside surface temperature, permits higher inside dew-points without condensation.

Whenever one begins speaking of per cent savings as mentioned previously, a certain minimum standard must be established and agreed to by all concerned. This provides a starting point from which these savings may be measured. It is my belief that such a minimum standard for building construction would include the equivalent of 1 in. of rigid insulation added to roof or top floor ceiling, a wall construction that would theoretically prevent inside surface temperature from falling below 60 F during cold weather (outside design temperature), and weatherstripping of doors and windows.

In the case of residences built to such standards, glass and wall losses account for approximately equal fractions of the total. The owner is then confronted with the problem of deciding whether an expenditure for storm windows or additional wall insulation would prove more profitable. Based on the average ratio of net wall area to window area, the coefficients for walls published in THE GUIDE 1938 and the data from this paper, the total extra cost for double glazing may be from 1⅓ to 2 times the total extra cost of wall construction and keep the added resistance to heat loss per dollar spent equal.

This relation may be expressed in the following equation:

$$\frac{Y}{X} = \frac{0.4\ A_g}{(U_w - U'_w)\ A_w}$$

Y = Extra cost for double glazing, dollars.

X = Extra cost for wall insulation (above minimum standard).

A_g = Area of glass, square feet.

A_w = Net area of wall, square feet.

U_w = Coefficient for minimum standard wall, Btu per hour per square foot per degree Fahrenheit.

U'_w = Coefficient for insulated wall, Btu per hour per square foot per degree Fahrenheit.

0.4 = Difference between single and double glazing, this paper. Btu per hour per square foot per degree Fahrenheit.

If the ratio of the actual costs exceeds that shown by this equation, then it would be a better investment to purchase the insulation. If it is less than indicated by the equation, then double glazing is the better buy.

The authors call attention to differences between their data and values published in THE GUIDE, indicating as other recent papers have also done that perhaps the recommended coefficients should be revised upwards. This question cannot be settled without giving consideration to what should be taken as the proper design velocity of outside air. It may be that sufficient data are already available, and perhaps a committee could be appointed to make recommendations for accepted good practice in heat loss calculations after it has studied the data.

W. C. RANDALL (WRITTEN): There are discrepancies between the test results of this paper and the data which were the bases of a previous paper.[19] In looking for the answer to such variations, certain test conditions seemed to offer the explanations.

From the diagram of the set-up, Fig. 1, it appears that the cooling coils are unguarded and exposed to direct radiation from the warmer surfaces, particularly from the outside surface of the glazed windows. Since the coils are colder than the air, direct radiation would occur, with increased heat transmission and without appreciably affecting the temperature of the air in the cold room. We have attempted to evaluate the direct radiation to the cold coils by the use of Stefan and Boltzmann's law with results that showed this to be of the magnitude of from 15 per cent to 34 per cent of the total heat flow and a logical conclusion seems to be that a figure of from 20 per cent to 25 per cent would be possible. That, if true, might explain the difference between the computed figures for glass alone and the higher test results for the glazed steel windows. In our tests we not only shielded the cold box but also the thermometers and the couples.

It does not seem that 0.66 mph on the outside and 2.18 mph on the inside represent anywhere near natural conditions. Still air, both inside and outside, and still air on the inside and 15-mile wind on the outside which was the basis of our test, made it easier to check against the present figures in THE GUIDE.

There is an incompleteness of test data and the combination of test data, computations and assumptions. For instance, on p. 477 "Those for the 36-light windows are less exact, as only three points were determined. These curves were fitted by the method of least distances." I would hesitate to draw curves, Fig. 6, with the points shown because they seem to fall in horizontal line groups with variations between the groups rather than on sloping curves increasing with mean temperature. In our tests we did not find such a trend. For 19 sets of tests on all kinds of windows we found an increase in but two, a decrease in nine, and a varying trend in eight.

I do not consider the double glazing detail practical or one that would be recommended by the steel window industry. Double glazing of steel windows is of doubtful value, particularly with high inside relative humidity. At least we would not recommend aluminum spacer strips. The outside lights would probably be larger than the inside lights; the outside lights would, no doubt, be removable so that washing could be done between the lights, and the vents to the outside would have protection against breathing of dust and dirt into the space between the lights.

H. C. DICKINSON[20] (WRITTEN): I am somewhat doubtful about the use of a Kata thermometer within 1 in. of a window at a considerable difference in temperature from the air surrounding the Kata. The authors may be optimistic about the precision of the air velocities reported. I must confess that I am not at present able to suggest a better method for obtaining absolute air velocities at the same position. The practical use of the data on the effect of air velocity 1 in. from the pane is difficult to perceive since the wind velocity around any given house that will result in a certain air velocity 1 in. from a window pane is unknown.

[19] Room Surface Temperature of Glass in Windows, by J. E. Emswiler and W. C. Randall. (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 181).

[20] National Bureau of Standards, Washington, D. C.

I have no particular comment to offer on this paper other than to point out that the over-all heat transfer coefficient of a window (single or double) must be a function of both mean temperature and temperature difference, considered as independent variables. In this paper, the warm side is kept at 70 F, and both mean temperature and temperature difference varied simultaneously by varying the cold side temperature. The data are expressed as a function of mean temperature, which is, of course, perfectly definite in this case, since the warm side temperature was always the same. The data on temperature distribution should have been tabulated, since it is extremely difficult to read off the charts.

As the problem appears to me it would be well to concentrate more on the separate elements of heat transfer, rather than to the extent the authors have done on the overall effects.

Two points in particular are mentioned in the paper as calling for more attention and I quite agree with this conclusion. The surface coefficients obviously play a predominant part in heat transmission through glass windows as the material itself is a relatively good conductor of heat in the thickness in which it is used. While much work has been done on the subject, relating temperature gradients to rates of air flow, the authors have not made as much use of existing data as they might have done. I believe it was suggested by Irving Langmuir a good many years ago that the heat transfer through an air layer next to a solid surface might be conveniently computed by assuming a dead air layer in contact with the surface, having a thickness which is a direct function of the air velocity at some arbitrarily selected location near the surface but not too near it. There are, of course, theoretical as well as practical limitations to this process but it is nevertheless a rational approximation to the correct solution, which is both simpler and perhaps more convenient and accurate than many of the devices which have been used. I believe its use might simplify the problem in respect to windows.

The second point has to do with condensation of moisture. Here it is obvious that the collection of moisture between the two panes of a double-glazed window depends on the relative permeability of the joints on the warm side and on the cold side. Clearly, if the warm side is tight enough and the cold side is ventilated enough, there will be no condensation. There is no question at all about the possibility of avoiding condensation inside double-glazed windows except under very unusual and temporary conditions if a proper design is worked out. A discussion of the matter on this basis might be more helpful to those who are trying to make use of the insulating value of double glazing.

AUTHORS' CLOSURE: If nothing more, the discussion certainly reveals a lively interest in a problem, the answer to which we hoped could be obtained with a few tests. Instead, the desirability of research was indicated. The outset of this closure seems the best place to reply to one commentator that the tests were undertaken to obtain data for which architects and engineers were asking. Otherwise we are taking up questions raised in the order that comments are presented.

We agree with Mr. McDermott that, if wood spacers did not swell, decay or cause staining, theoretically at least, they would be preferable to metal spacers. The Kata thermometer readings that he asks about were taken with ordinary room temperatures on both sides of the window and with fans adjusted to produce as nearly uniform air movement as possible.

With respect to Professor Humphrey's comments, we soon discovered air movement to be much more of a factor than anticipated. As our budget did not permit measuring loss at several combinations of air velocities, we decided to work with as low air movement as possible to pertain without stratification, and then later to determine the effect of air movement. Unfortunately, due to depressions and other

factors, this is yet to be done. We did, however, decide to define in some manner under what conditions of air movement the tests were conducted. In studying the literature one is struck with the fact that attendant conditions of air movement are either not mentioned, or only vaguely described. It would not be surprising if most of the differences in results obtained by various investigators could be accounted for by differences in air movement.

We believe the foregoing covers Mr. Roger's comments also.

In connection with Professor Queer's comments, in those cases where indoor condensation might be troublesome with double-glazed windows, it would be much more troublesome with single glazing. No attempt was made to make a thorough study of condensation. The data given were obtained as a result of our endeavor to record all pertinent facts with respect to conditions under which the tests were conducted. As in the case of air movement, we have often found the literature vague with respect to humidity conditions. We now agree with Professor Queer that height of air space, as well as muntin area, was undoubtedly a factor in increasing heat loss through the smaller light window.

No attempt was made to measure true surface temperatures because of the difficulties involved. If some one can tell us how *true* surface temperatures of glass can be determined, we wish he would do so. We did place thermocouple junctions in contact with glass even though we realized that they would not indicate true surface temperatures. The maximum difference between calculated surface temperatures and the reading of thermocouples in contact with glass was about 5 deg F.

The refrigeration coils were not shielded. This suggests a possible source of error although in a recent test of a tile panel conducted with shielded and unshielded coils, no difference in heat transfer was found.

We appreciate Professor Rowley's work in showing that calculated and measured heat losses through the windows are in substantial agreement.

Concerning Mr. Tenkonohy's remarks, while we recognized the desirability of tests with 15 mph air movements, we did not for reasons already explained, conduct such. We did, out of curiosity, make a few runs at about 7.5 mph air movement on the cold side just to obtain some conception of the magnitude of the effect of air movement. The following table indicates that the losses in these runs were higher than Mr. Tenkonohy's estimate.

BTU PER HOUR PER SQUARE FOOT PER DEGREE FAHRENHEIT DIFFERENCE

Test Condition	(2.18–7.58 MPH) Measured Losses	(0–15 MPH) Mr. Tenkonohy's Figures
9 Lt. W.lw. S. G.....	1.74	1.26
9 Lt. Wdw. D. G.....	0.86	0.675
36 Lt. Wdw. S. G.....	1.78	1.40
36 Lt. Wdw. D. G.....	0.99	0.785

Based upon calculations for glass alone, using film coefficients found in THE GUIDE, the over-all coefficient for 2.18 — 7.58 mph conditions would actually be 10 per cent greater than for 0 — 15 mph conditions.

We are glad to have Colonel Danielson add emphasis to the importance of air movement and condensation. The last paragraph of Mr. Dickenson's comments sums up admirably the facts with respect to the latter.

We agree with Mr. Rockwell that it should have been pointed out in the paper that the heat losses referred to were window losses, not total building losses.

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Respecting Mr. Randall's comments regarding radiation, the logical thing to have done would have been to repeat at least one test with refrigeration coils shielded. This was not done at the time but has been done since, with a tile wall, not a window however, and no difference in heat loss was found. We have also, since conducting the tests herein reported, made measurements of coil temperatures and computed, that the radiation loss under the most severe conditions might have been 10 per cent provided certain assumptions were correct. By making various assumptions, we found it possible to arrive at almost any computed loss. Mr. Randall's comments suggest a possible window heat loss that we do not recall ever seeing discussed, namely, radiation to outdoor objects within the *view* of windows.

The sash used were furnished us by a large sash manufacturer who knew for what purpose and how they were to be used. The method of double glazing was decided upon with a view to its practical application in industrial buildings with large areas of glass.

We have already explained why and how the Kata thermometer readings referred to by Mr. Dickenson were made. One inch for distance from the glass was selected arbitrarily as being as good as any other one distance for defining test conditions. It will at least give some future investigator who may read this paper in his study of the literature some idea as to the conditions of air movement. We made no attempt to break the data down into the elements of heat transfer as Mr. Dickenson suggests since the designing engineer is interested primarily in over-all heat loss.

In conclusion, the data show that it is physically possible to double glaze certain kinds of installations of industrial sash and thereby reduce heat losses. Whether it is economically feasible to do so is another matter. We believe it should at least be given careful consideration.

No. 1097

MEASUREMENT OF HEAT TRANSFER BY FREE CONVECTION FROM CYLINDRICAL BODIES BY THE SCHLIEREN METHOD

By L. M. K. BOELTER * (*NON-MEMBER*) AND V. H. CHERRY ** (*MEMBER*)
BERKELEY, CALIF.

THE rate of heat transfer from the surface of a body surrounded by a diathermanous fluid is due to convection and radiation to the surroundings. Often in experimental technique, the free convection rate is calculated as a difference between the heat rate input and the rate of radiant energy transfer. If the latter term is not measured separately, the use of an estimated emissivity may introduce a large error.

There are several means by which the error in the determination of heat transfer by free convection due to the uncertainties of the emissive power of a surface may be reduced or eliminated.

1. The rate of heat transfer from solids to liquids by convection is so large that the radiation is negligible. By the correlation with generalized variables (dimensionless moduli) the data determined in this manner may be extrapolated to other fluids.

2. The radiation from small wires may be negligible in comparison with the convective heat losses.

3. The solid may be given a highly polished surface so that an error of 50 per cent in the emissive power may amount to an error of only 5 per cent in the determined convective losses.

4. By means of the Schlieren method developed by Schmidt,[1] the temperature gradient in the fluid near the surface of the solid may be determined. The temperature gradient multiplied by the thermal conductivity of the fluid results in the convective heat rate per unit area. The disadvantages of the method are:

a. The thermal conductivities of fluids are not accurately known. Variations of 10 per cent in published data may be noted.[2]

b. The method is only applicable to surfaces of such shape as may be created by a straight generatrix along which the temperature is uniform. The stagnant film of fluid on the surface must be between 0.02 in. and 0.10 in. in thickness.

c. These disadvantages are offset by the supplementary data which may be obtained. In addition to the total heat rate by convection, the film conductance as a function of distance along the directrix of a cylinder may be determined. This is an aid, for example, in computing the thickness of ice and frost on the expansion coils

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[1] Schlierenaufnahmen des Temperaturfeldes in der Nähe wärmeabgebender Körper, by E. Schmidt (*Forschung*, Vol. 3, No. 4, 1932, p. 181-189).
[2] Heat Transmission, by W. H. McAdams (McGraw-Hill, 1933).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by John James.

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of a refrigeration system.[3] The resulting photographic data are also an aid in the visualization of the free convection process.[4]

Data obtained by means of the Schlieren method are employed in this paper to determine the effect of some simply shaped surroundings upon the convective heat rate from a horizontal circular cylinder.

In order to generate confidence in the Schlieren experimental technique, the

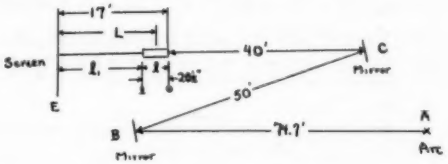

FIG. 1. SCHEMATIC SKETCH OF APPARATUS TO DETERMINE HEAT EXCHANGE RATE FROM A HORIZONTAL CYLINDER TO AIR

measured film conductances by free convection in air from a horizontal cylinder far from surroundings are compared with data from the literature.

EXPERIMENTAL TECHNIQUE

The rate of convective heat transmission may be shown to depend upon the slope of a beam of light as it leaves the thermal field near surface.[5] The light beam is so directed as to be tangent to the surface at the entrance to the thermal field. The beam slope is determined experimentally.

Fig. 1 is a schematic sketch of the apparatus arrangement which was used to determine the heat exchange rate from a horizontal cylinder to air.

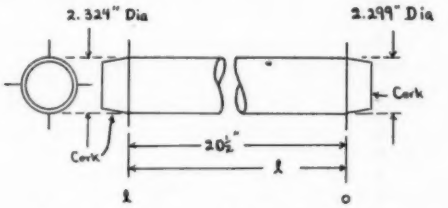

FIG. 2. DIAGRAM OF CYLINDER SHOWING PROJECTING NEEDLES

An arc lamp was used as a source of light. To insure that all of the light beams were parallel to the surface of the cylinder, the cylinder was slightly

[3] Der Wärmeübergang an Rohren bei freier Strömung unter Berucksichtigung der Bildung von Schwitzwasser und Reif, by W. Piening (Gesundheits Ingenieur, No. 42, Oct. 1933).
[4] An Optical Method for Measuring Temperature Distribution and Convective Heat Transfer, by R. B. Kennard (*Bureau of Standards Journal of Research*, Vol. 8, June, 1932, No. 6, p. 787).
[5] Photographic Records of Heat Transfer, by S. Ruppricht (*Refrigerating Engineering*, Vol. 25-26, 1928).

tapered. To reduce the taper, and so that the arc lamp might be considered substantially a point source it was necessary that the cylinder be located at a considerable distance from the lamp. This result was accomplished by reflecting the light from plane mirrors as shown in Fig. 1. The distance from the lamp to the O end of the cylinder was 164.7 ft. The diameter of the cylinder varied from 2.299 in. on the O end to 2.324 in. on the l end. The deflection of the light beam initially tangent to the cylinder on the O end was determined by exposing sensitized paper on a screen E located 17 ft from the O end of

FIG. 3. VIEW OF CYLINDER ARRANGED FOR ADJUSTMENTS

the cylinder. The cylinder was chromium plated to reduce the surface emissivity and to preserve the surface.

The angle which the light beams made with the center of the cylinder was $2.30/2 \cdot 12 \cdot 164.7 = 0.000582$ radians. If the cylinder had not been tapered the light beams would have been $0.000582 \times 20\frac{1}{2} = 0.0121$ in. from the cold cylinder surface at the l end. The cylinder was so tapered that the light beam was tangent for the complete length of the cold cylinder. The diameter of shadow of the cold cylinder should have been

$$\frac{164.7 + 17.0}{164.7} \times 2.30 = 2.54 \text{ in.}$$

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The measured shadow of the cold cylinder was slightly larger (0.06 in.). This effect was attributed to diffraction and to the penumbra due to the finite size of the source. The computed cold shadow size was used in the calculations.

The cylinder was aligned with the light beam by means of projecting needles on each end, as shown in Figs. 2, 3 and 4. The cylinder was considered to be aligned when the shadows of corresponding projecting needles coincided on screen *E* (Fig. 1). A support for the cylinder was so arranged that final alignment could be made by means of leveling screws.

The center of the cold shadow was located at the intersection of lines through the images of the needles.

As shown in the appendix, the slope of a light beam (initially tangent) as it left the cylinder was determined by the deflection from the shadow cast

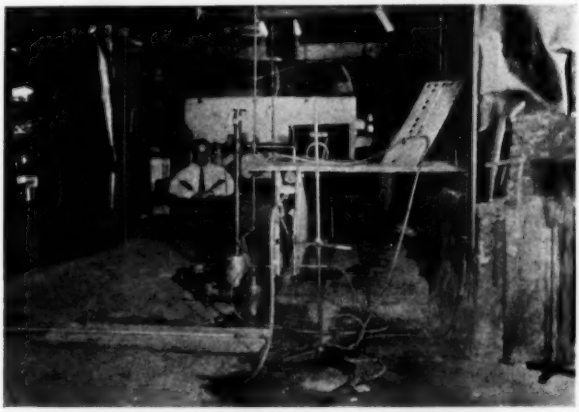

FIG 4. GENERAL VIEW OF TEST APPARATUS

by the cold cylinder on the screen. The deflection was considered to be the distance from the computed shadow of the cold cylinder to the line of impingement of the outermost beam of light in the shadow from the hot cylinder. The deflection of the light beam was recorded by the exposure of sensitized paper at the screen *E*. (Examples of photographs obtained are shown in Figs. 5 and 6.)

For uniformity of temperature over the surface the cylinder was made from a piece of solid shafting with a ½-in. diameter hole drilled through the center in which an electrical heater was placed.

To show the effect of certain simply shaped surroundings, flat, water-cooled walls at various distances from the cylinder were arranged to form:

1. A vertical wall parallel to the cylinder.
2. A horizontal ceiling.
3. An intersection of a wall and ceiling parallel to the cylinder.
4. A tunnel, square in cross-section, located concentrically about the cylinder.

All flat surfaces were 3 ft long in a direction parallel to the cylinder. The

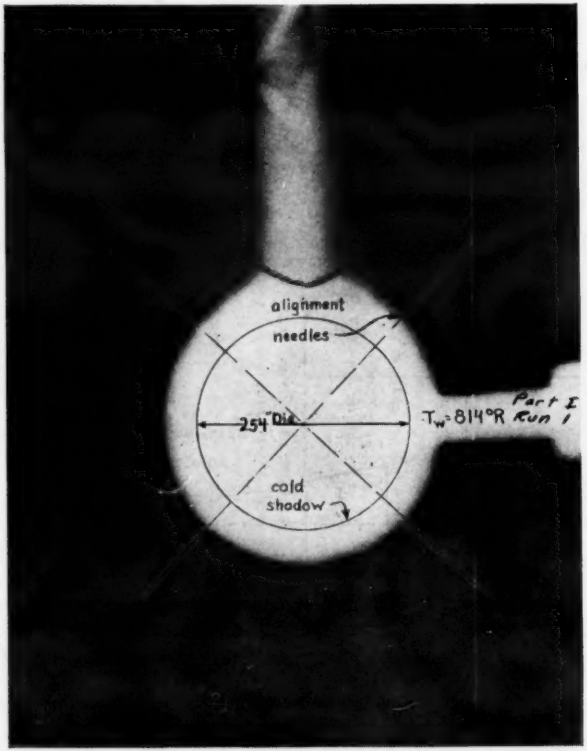

FIG. 5. PHOTOGRAPHIC RECORD SHOWING DEFLECTION OF LIGHT BEAM FOR RUN 1

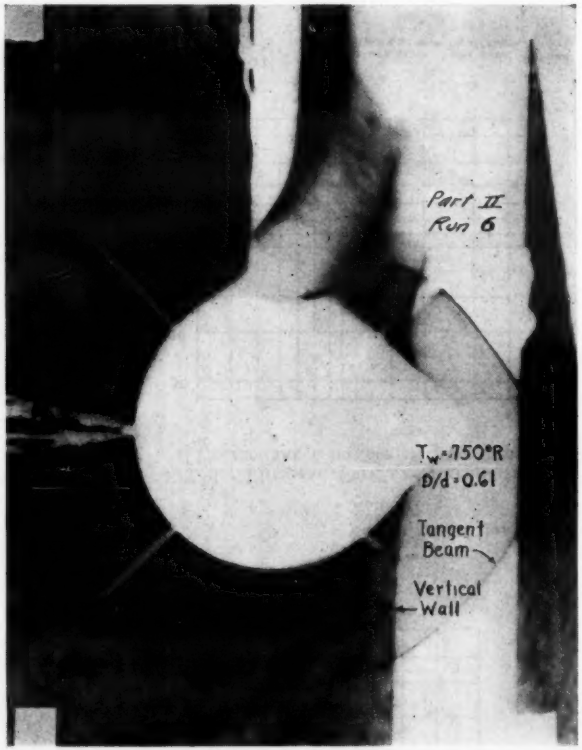

FIG. 6. PHOTOGRAPHIC RECORD SHOWING DEFLECTION OF LIGHT BEAM FOR RUN 6

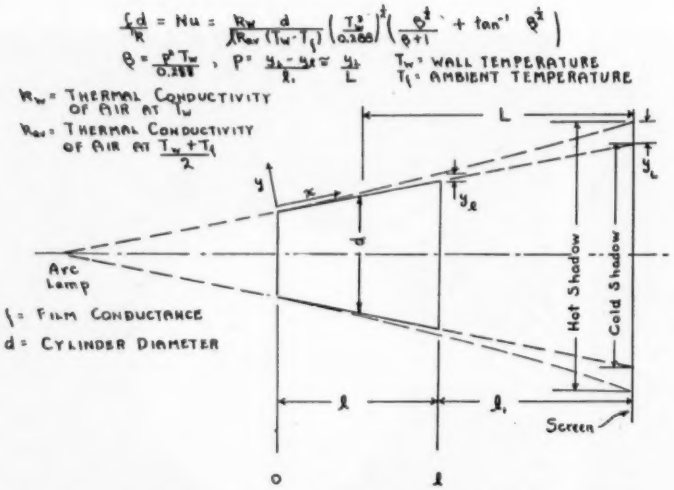

FIG. 7. (*above*) DIAGRAMMATIC SKETCH OF CYLINDER FOR DEVELOPING COMPUTATION OF CONVECTIVE RATE OF HEAT LOSS

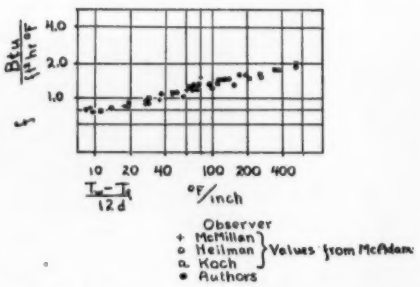

FIG. 8. THERMAL FREE CONVECTION FROM A HORIZONTAL CYLINDER IN FREE SPACE

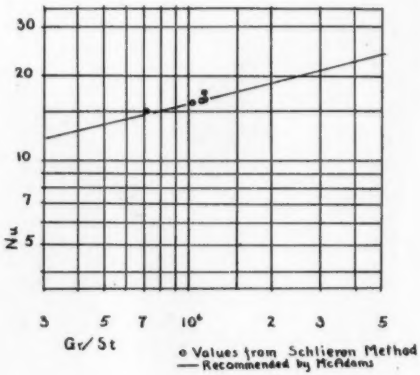

FIG. 9. (*right*) COMPARISON OF FILM CONDUCTANCE VALUES IN TERMS OF *Nu* AND *Gr/St*

room temperature varied from 75 to 78 F. The wall temperatures were 5 to 8 F below the room temperature.

EVALUATION OF EXPERIMENTAL RESULTS

The equation employed for the computation of the convective rate of heat loss, as indicated in Fig. 7 and appendix, was:

$$fd/k_{av} = Nu = \frac{k_w d}{l k_{av}(T_w - T_f)}\left(\frac{T_w^3}{0.288}\right)^{\frac{1}{4}}\left(\frac{\beta^{\frac{1}{2}}}{\beta+1} + tan^{-1}\beta^{\frac{1}{2}}\right) \qquad (1)$$

where

β $= p^2 T_w/0.288$.

p $= \frac{y_L - y_l}{l_1} \cong \frac{y_L}{L}$ = slope of light beam as it leaves the cylinder.

T_w = temperature of cylinder, degrees Fahrenheit absolute.

T_f = temperature of ambient air, room temperature, degrees Fahrenheit absolute.

y_L = deflection of tangent light beam at screen E, feet.

L = distance from center of cylinder to screen, feet.

y_l = deflection of light beam as it leaves the cylinder, feet.

l = length of cylinder, feet.

l_1 $= L - l/2$.

k_w = thermal conductivity of air at temperature, T_w, Btu per (square foot) (hour) (degree Fahrenheit per foot).

k_{av} = thermal conductivity of air at the average temperature, $\frac{T_w + T_f}{2}$, Btu per (square foot) (hour) (degree Fahrenheit per foot).

d = cylinder diameter, feet.

f = convective film conductance per unit area, Btu per (square foot) (hour) (degree Fahrenheit).

The equation is based upon the following conditions:

1. For the path through which the tangent light beam travels, the air temperature is a linear function of the distance measured perpendicular to the cylinder wall.
2. The index of refraction of air is given by the equation

$$n = 1 + \frac{0.144}{T}\frac{b}{760}$$

where

b = barometric pressure in millimeters of mercury.

T = air temperature, degrees Fahrenheit absolute.

3. The diffraction and penumbra effects are small.
4. The end conditions do not affect the light beam.

Equation (1) differs from that given by Schmidt[6] in that

$\left(\frac{\beta^{\frac{1}{2}}}{\beta+1} + tan^{-1}\beta^{\frac{1}{2}}\right)$ replaces $2\beta^{\frac{1}{2}}$.

The ratio $2\beta^{\frac{1}{2}} \Big/ \left(\frac{\beta^{\frac{1}{2}}}{\beta+1} + tan^{-1}\beta^{\frac{1}{2}}\right)$ may be expressed approximately as $\eta = 1 + 0.62\,\beta$ (2)

[6] Loc. Cit. See Note 1.

In the experimental work summarized below η may be as large as 1.2.

EXPERIMENTAL RESULTS

To justify the assumptions made in the derivation of equation (1), the values of film conductances determined for a horizontal cylinder placed in free space by the Schlieren method are compared with those obtained by other methods. In Fig. 8 the film conductances given in Table 1 are shown as a function of

$$\frac{\text{temperature difference}}{\text{cylinder diameter}}$$

and agreement is well within the accuracy of the data.

In Fig. 9 the data are compared with recommended values in terms of the generalized variables Nu and Gr/St,

where

Gr = Grashof's modulus = $\dfrac{gd^3\rho^2(T_w - T_f)}{\mu^2 T_{av}}$

St = Stanton's modulus = 1/Prandtl's modulus = $\dfrac{k_{av}}{\mu g c_p}$

$T_{av} = \dfrac{T_w + T_f}{2}$

μ = viscosity of fluid, pound second per square foot.
ρ = mass density of fluid, pound (second)^2 per (foot)^4.
k_{av} = thermal conductivity of fluid, Btu per (square foot) (second) (degree Fahrenheit per foot).
c_p = specific heat of fluid at constant pressure, Btu per (pound) (degree Fahrenheit).
g = acceleration due to gravity, feet per second squared.

All fluid properties were determined at the mean film temperature and were taken from the International Critical Tables.

The experimental data agree well with the curve recommended by McAdams.[7]

The effects of four geometrically simple boundary conditions, given in Table 2, are indicated graphically in Fig. 10. The ratio of film conductance, with the cylinder at any distance from the walls, to that with the walls far from the cylinder, is shown as a function of the dimensionless ratio, the distance of cylinder center from wall/diameter of cylinder. The value of T_f in Grashof's Modulus and in equation (1) was taken as the room temperature. The correlations may not apply outside of the range of data investigated.

ACKNOWLEDGMENT

The authors wish to express their appreciation of the work of W. B. Chiswell, T. B. Klakring, W. Stahl, A. R. Joyce, F. Agens, and J. C. Broach,

[7] Loc. Cit. See Note 2.

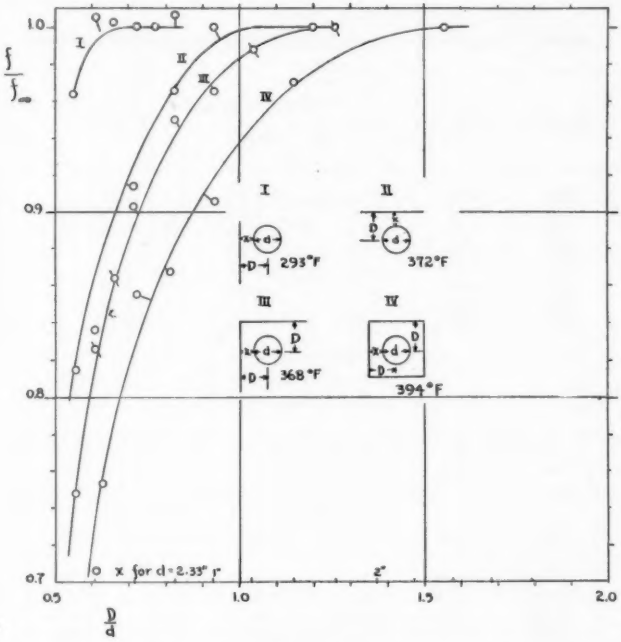

FIG. 10. HEAT TRANSFER BY FREE CONVECTION FROM HORIZONTAL CYLINDERS AS A FUNCTION OF DISTANCE FROM SURFACES VARIOUSLY ARRANGED

former graduate students, and D. N. Harris and T. R. Griffin, graduates of the University of California, Department of Mechanical Engineering.

These men aided in the development of the experimental technique and obtained the laboratory data.

APPENDIX

1. The index of refraction n of a medium may be defined by

$$Vn = C \tag{3}$$

where C is the velocity of light in a vacuum and V the velocity of light in the medium.

In Fig. 11 let R be the radius of curvature of a beam of light passing the surface $y = O$. The velocity of light at radius R is V and at $R + dR$, $V + dV$. In time $d\theta$ the beam at A travels a distance $Vd\theta$, while ray at B travels a distance $(V + dV)d\theta$.

By similarity of triangles $\frac{1}{R} = \frac{1}{V}\frac{dV}{dR}$ (4)

TABLE 1. FILM CONDUCTANCE VALUES FOR HORIZONTAL CYLINDER

RUN	T_w, DEG F ABS.	T_f, DEG F ABS.	Nu_{av}	f BTU PER (HOUR) (SQ FT) (DEG F)	$(Gr/St) \times 10^{-6}$	$\frac{(T_w - T_f)}{12d}$ DEG F PER INCH
1	814	537	16.7	1.43	1.17	119
2	700	535	16.0	1.29	1.04	71
3	759	535	16.3	1.35	1.13	96
4	800	535	16.7	1.43	1.17	114
5	856	535	16.8	1.47	1.18	138
6	935	535	17.5	1.60	1.15	172
7	652	535	15.0	1.15	0.72	50
8	831	535	16.7	1.46	1.17	127

TABLE 2. EFFECTS OF FOUR GEOMETRICALLY SIMPLE BOUNDARY CONDITIONS

VERTICAL WALL		CEILING		CORNER		TUNNEL	
D/d	f/f_∞	D/d	f/f_∞	D/d	f/f_∞	D/d	f/f_∞
0.93	0.992	1.20	1.0	1.26	1.0	1.56	1.0
0.83	1.008	0.93	1.0	1.04	0.988	1.15	0.970
0.77	1.000	0.83	0.963	0.93	0.975	0.93	0.905
0.72	1.001	0.72	0.915	0.83	0.949	0.83	0.867
0.66	1.003	0.66	0.858	0.72	0.903	0.72	0.857
0.61	1.005	0.61	0.837	0.66	0.864	0.67	0.752
0.55	0.963	0.56	0.815	0.61	0.826	0.61	0.706
				0.56	0.748		

TABLE 3. TABULATION OF CONVECTIVE HEAT LOSS—CORRECTION FACTOR (η)

β	$\phi_2(\beta)$	η	β	$\phi_2(\beta)$	η
0.0005	0.0448	1.000	0.04	0.390	1.026
0.001	0.0632	1.000	0.05	0.433	1.033
0.002	0.0892	1.002	0.10	0.594	1.065
0.003	0.1094	1.003	0.15	0.706	1.095
0.004	0.1262	1.003	0.20	0.793	1.126
0.005	0.1410	1.003	0.25	0.864	1.157
0.006	0.1543	1.004	0.30	0.923	1.188
0.007	0.1666	1.004	0.35	0.973	1.216
0.009	0.1885	1.006	0.40	1.016	1.243
0.01	0.1987	1.007	0.45	1.054	1.275
0.02	0.279	1.014	0.50	1.088	1.300
0.03	0.340	1.019	0.55	1.117	1.328
			0.60	1.143	1.355

Combining equations 3 and 4 gives

$$\frac{1}{R} = \frac{-1}{n}\frac{dn}{dt}\frac{dt}{dR} \tag{5}$$

If the ray is approximately parallel to the surface $y = O$,

$$\frac{1}{R} = \frac{d^2y}{dx^2} = \frac{+1}{n}\frac{dn}{dt}\frac{dt}{dy} = \frac{+m}{n}\frac{dn}{dt} \tag{6}$$

where, y = distance perpendicular to the surface,
x = distance parallel to the surface,

$m = \frac{dt}{dy}$, t = temperature.

For air, $n = 1 + \frac{0.144}{T}\frac{b}{760} = 1 + \frac{A}{T}$ (7)

b = barometric pressure in millimeters mercury.
T = temperature, degrees Fahrenheit absolute.

$$A = 0.144\,\frac{b}{760}$$

Since n is within 0.03 per cent of being unity,

$$\frac{1}{n}\frac{dn}{dt} = \frac{-A}{T^2} \tag{8}$$

Then $\frac{d^2y}{dx^2} = \frac{-mA}{T^2}$ (9)

The boundary conditions for the integration are,

(a) $x = O$, $\frac{dy}{dx} = O$, $y = O$

(b) $x = l$, $\frac{dy}{dx} = p$, $y = y_l$

For convenience in computation, define the dimensionless quantity β as $\frac{p^2T_w}{2A}$,

T_w = surface temperature degrees Fahrenheit absolute.

2. If the cylinder is short, the surface temperature low, and the transfer factor low, the deflection, y_l, is small. Then T in equation (9) may be considered constant. With $T = T_w$, integration of equation (9) yields

$$\frac{dy}{dx} = \frac{-Amx}{T_w^2} \tag{10}$$

$$p = \frac{-Aml}{T_w^2} \tag{11}$$

$$\text{or } m = \frac{-pT_w^2}{lA}$$

$$m = \frac{-1}{l}\sqrt{\frac{T_w^2}{2A}}\,2\sqrt{\beta} \tag{12}$$

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$$= \frac{-1}{l}\sqrt{\frac{T^3_w}{2A}}\ \phi_1(\beta) \tag{12a}$$

With the thermal conductivity determined at the mean temperature of the tube and of the ambient air, and with the tube diameter as the significant length, Nusselt's modulus is

$$Nu = \frac{k_w d}{k_{av}(T_w - T_l)l}\sqrt{\frac{T^3_w}{0.288}}\ \phi_1(\beta) \tag{13}$$

3. For higher temperatures and longer cylinders the assumption $T = T_w$ in equa-

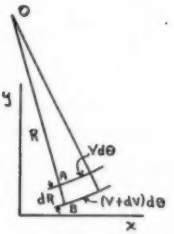

FIG. 11. DIAGRAM OF LIGHT BEAM TRAVEL

tion (9) is not valid. The temperature may be considered a linear function of distance out to 4 or 5 millimeters from the cylinder wall.[8]

Let $T = T_w + my$ in equation (9)

and $$\frac{d^2y}{dx^2} = \frac{-Am}{(T_w + my)^3} \tag{14}$$

Integrate with the limits

$$\frac{dy}{dx} = O, \text{ for } x = O,\ y = O, \text{ and } \frac{dy}{dx} = \frac{dy}{dx} \text{ at } x = x,$$

which yields $$\frac{dy}{dx} = \sqrt{\frac{2A}{T_w}}\ \sqrt{\frac{1}{1 + my/T_w} - 1} \tag{15}$$

Integration of (15) between the limits $y = O$ to y_1, $x = O$ to l yields

$$m = \frac{-T_w}{l\sqrt{\frac{2A}{T_w}}}\left[z^{\frac{1}{2}}(1-z)^{\frac{1}{2}} + tan^{-1}\left(\frac{z}{1-z}\right)^{\frac{1}{2}}\right] \tag{16}$$

where $z = \dfrac{-my_1}{T_w}$

[8] The Transmission of Heat by Radiation and Convection by E. Griffiths and A. H. Davis (Dept. of Scientific and Industrial Research, Food Investigation Special Report, No. 9, 1931, rev. ed., H. M. Stationery Office, London).

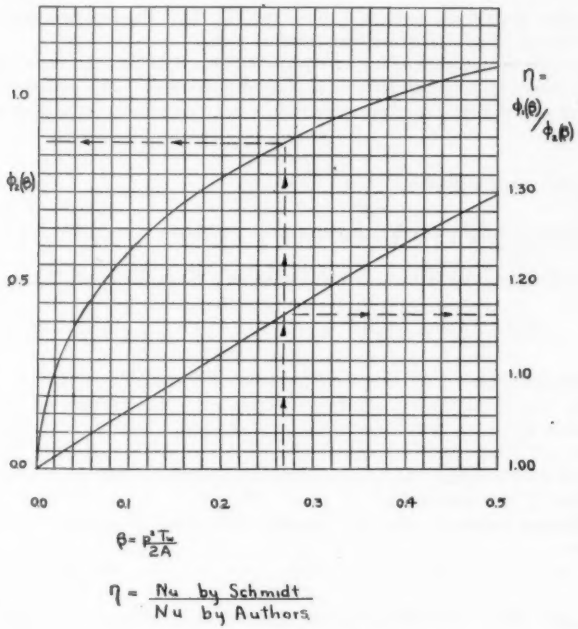

FIG. 12. MAGNITUDES OF ϕ_2 (β) AND η

From equation (15)

$$p = \sqrt{\frac{2A}{T_w}} \sqrt{\frac{1}{1-z} - 1} \tag{17}$$

From equation (17)

$$z = 1 - \frac{1}{\beta + 1} \tag{18}$$

With equation (18) in equation (16)

$$m = -\frac{1}{l}\sqrt{\frac{T_w^3}{2A}}\left(\frac{\beta^{\frac{1}{2}}}{\beta+1} + tan^{-1}\,\beta^{\frac{1}{2}}\right) \tag{19}$$

$$m = -\frac{1}{l}\sqrt{\frac{T_w^3}{2A}}\,\phi_2(\beta) \tag{19a}$$

Thus

$$f = \frac{-k_w m}{(T_w - T_l)} = \frac{k_w\sqrt{T_w}\phi_2(\beta)}{(1 - T_l/T_w)l\sqrt{2A}} \tag{20}$$

and

$$Nu = \frac{k_w d}{k_{av}(T_w - T_l)l}\sqrt{\frac{T_w^3}{0.288}}\,\phi_2(\beta) \tag{21}$$

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The values of $\phi_2(\beta)$ are given in Table 3 and Fig. 12. The function $\phi_2(\beta)$ approaches the function $\phi_1(\beta)$ as a limit as β approaches zero.

The ratio η of equation (12a) to equation (19a) indicates the error in employing equation (12a) for the determination of Nusselt's modulus. The values of η are shown in Table 3 and in Fig. 12. Approximately, η is a linear function of β and may be expressed by

$$\eta = 1 + 0.62\,\beta \tag{22}$$

As a close approximation,

$$Nu \cong \frac{k_w d}{k_{av}(T_w - T_l)l}\sqrt{\frac{T_w^3}{0.288}}\,\phi_1(\beta)\Big/(1 + 0.62\,\beta)$$

In the computation by either method, the slope p must be known. Exactly,

$$p = \frac{y_L - y_1}{l_1} \tag{23}$$

As an approximation, with $T = T_w$, equations (10) and (11) yield

$$y_1 \cong \frac{p}{2}l \tag{24}$$

Since y_1 is small compared to y_L, a small error in y_1 will cause a small error in the computation of p by equation (23).

Substituting equation (24) into equation (23) yields (Fig. 4)

$$p = \frac{y_L}{l_1 + l/2} = \frac{y_L}{L} \tag{25}$$

Equation (25) was employed for the determination of slope, p. The error due to this approximation may be two or three per cent. The actual value of p may be found by trial and error methods if greater accuracy is desired.

No. 1098

RADIATION AND CONVECTION FROM SURFACES IN VARIOUS POSITIONS

By GORDON B. WILKES * AND CARL M. F. PETERSON ** (*MEMBERS*)
CAMBRIDGE, MASS.

WHEN the rate of heat transmission through a wall or a roof is calculated, there is always a surface coefficient for *still* air to be considered. For ordinary surfaces such as wood, plaster, brick, etc., the value of 1.65 Btu per hour per square foot per degree Fahrenheit temperature difference is commonly used as given in the HEATING, VENTILATING AND AIR CONDITIONING GUIDE 1938 and similar publications. This value is used regardless of the position of the surface and the temperature difference. Fortunately, in the case of building walls, this surface coefficient is of minor importance, representing approximately 16 per cent of the total resistance to heat flow in an uninsulated frame wall and only about 6 per cent in a well-insulated wall. Problems frequently occur, however, where the surface coefficient is an important factor. The surface resistance of a single glass window accounts for nearly 100 per cent of the total resistance to heat flow, the thickness and thermal conductivity of the glass being negligible factors. Even in the case of a flat concrete roof slab, 2 in. thick, the surface resistance on the inside is responsible for roughly 50 per cent of the total.

In view of the occasional importance of surface coefficients, it was decided to determine how these coefficients for *still* air varied with position and temperature differences found in buildings. Little attempt has been made by previous investigators of surface coefficients to, first, separate radiation from convection; second, study the effect of position; and third, use large surfaces of 10 or more square feet. Griffiths and Davis [1] used a plate from 3 to 4 ft square and varied its position but used temperature differences considerably in excess of those in the present investigation.

The heat loss from a surface is due primarily to radiation and convection. The temperature difference ordinarily used in calculating a surface coefficient is based on the temperature of the surface and the temperature of the air two or more inches away from the surface. This procedure is correct for con-

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** Instructor in Heat Measurements, Massachusetts Institute of Technology.
[1] The Transmission of Heat by Radiation and Convection, by Griffiths and Davis (Food Investigation Board, Special Report No. 9, 1922, British).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by C. M. F. Peterson.

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vection losses only. For temperatures that are ordinarily found in building walls, it has been fairly well established that the mean temperature has little effect on the rate of heat loss by convection but convection increases with the temperature difference raised to some power greater than unity. The effect of position is a very important factor, heat being transferred by convection nearly three times as readily if the position is horizontal with heat flowing upward, than if the heat is flowing downward. Heat transfer by radiation, however, is independent of the ambient air temperature but depends upon the temperature of the surface of the heater plate and the surface temperature of the surrounding enclosure. The determination of surface coefficients by dividing the total (convection plus radiation) heat loss by the temperature difference between the surface and the ambient air would only be correct if the surrounding surfaces were the same temperature as the air. Frequently these

TABLE 1—VARIATION IN SURFACE COEFFICIENT WITH SURROUNDING SURFACE AT DIFFERENT TEMPERATURES

SURROUNDING SURFACE TEMPERATURE	75 F	70 F	69 F	60 F	50 F
Convection—Btu, per hour per square foot..	6.6	6.6	6.6	6.6	6.6
Radiation—Btu, per hour per square foot....	4.4	8.6	9.6	17.0	24.9
Total—Btu, per hour per square foot.....	11.0	15.2	16.2	23.6	31.5
Surface coefficient—surface and air temperature only.....	1.10	1.52	1.62	2.36	3.15

are not at the same temperature and since radiation may be responsible for a large percentage of the total loss, considerable error may be introduced.

The variation in the surface coefficient with the surrounding surface at different temperatures is shown in Table 1 if only the temperatures of the heater surface and the ambient air are used for calculation. The values apply only to a vertical surface at 80 F, the ambient air temperature at 70 F, and the effective emissivity equal to 0.83.

With the heater plate at 80 F, and the ambient air at 70 F, the error will be nearly 7 per cent if the surrounding surfaces are only 1 F cooler than the air. This error increases to 55 per cent if the surrounding surfaces are 10 F cooler than the ambient air.

In the case of ordinary surfaces with high emissivity, such as wood, plaster, paint, glass, etc., the surrounding surface temperature is of more importance than the ambient air temperature. The surface coefficient would be more nearly correct if the total heat loss were divided by the difference in temperature of the surfaces rather than the difference between the heated surface and the ambient air.

The previous example merely emphasizes the necessity of separating radiation from convection when determining surface coefficients.

The essential parts of the apparatus used in the measurement of surface coefficients have been previously described.[2] Some minor changes were made,

[2] Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 351.)

however, that materially improved the apparatus for this particular work. The original tempering tank, consisting of a coil of copper tubing immersed in a tank of water, was refrigerated by means of a one-half ton refrigerating unit and arrangements were made at the same time to heat the tank with gas. The refrigerating machine operated continuously and the temperature of the water was determined by the amount of gas heat. This produced a uniform low temperature that was not possible with the usual controls on the refrigerating unit alone. An aluminum sheet covered the entire calorimeter, making an air space approximately 4 in. thick, and thermocouples were attached so that the tem-

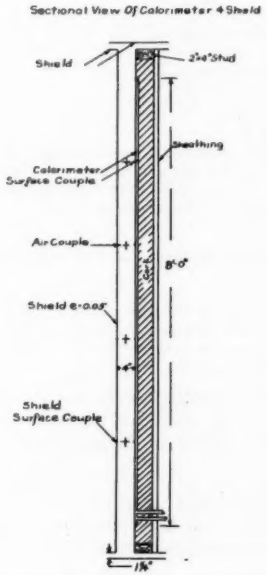

FIG. 1. SECTIONAL VIEW OF CALORIMETER AND SHIELD

perature of this surface could be measured accurately. The edges of this air space were open but baffled so as to minimize draughts. Fig. 1 shows the general arrangement of the air passages. Shielded thermocouples were placed midway in this air space, about 2 in. from each surface so that the ambient air temperature could be measured. A housing was constructed around the complete apparatus and the temperature of the air inside maintained a few degrees above the outside room temperature. Electrical heating was used in connection with a thermostat so that this temperature did not vary more than a small fraction of 1 deg during the tests. The heat flow and other temperature measurements were made in similar manner to the earlier experiments.

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Tests were made in five different positions, namely, horizontal, heat flow upward; 45 deg heat flow upward; vertical; 45 deg heat flow downward; and horizontal, heat flow downward. In each of the positions, runs were made at three distinct temperature differences.

In all cases, radiation was kept at a minimum by having one aluminum surface so that the convection loss could be more accurately measured. The values for ordinary surfaces with an effective emissivity of 0.83 were calculated using the convection loss as determined experimentally and calculating the radiation loss by the well-known formula,

$$q_r = e_{eff} A\sigma (T_2^4 - T_1^4)$$

where

q_r = radiation loss, Btu per hour

e_{eff} = effective emissivity $= \dfrac{1}{\dfrac{1}{e_1} + \dfrac{1}{e_2} - 1}$

A = area in square feet.
T_2 = temperature of warmer surface in degrees Fahrenheit absolute.
T_1 = temperature of cooler surface in degrees Fahrenheit absolute.
σ = constant = 17.2×10^{-10} Btu per hour per square foot per degree Fahrenheit absolute[4].

The heat loss from a plane surface in various positions is shown in Fig. 2 for the following conditions:

Temperature of warmer surface	80 F
Temperature of cooler surface	70 F
Temperature of ambient air	70 F

The lower curve shows convection only versus position, the middle curve, radiation plus convection with effective emissivity equal to 0.05, while the upper curve shows radiation plus convection with the effective emissivity equal to 0.83.

With a surface of low emissivity, the heat loss is nearly three times as great when in a horizontal position heat flow upward as when the heat flow is downward, all other conditions remaining the same.

Table 2 shows the relative amount of radiation and convection in various positions when the air and the surrounding surfaces are at the same temperature as in Fig. 2.

TABLE 2—RELATIVE AMOUNT OF RADIATION AND CONVECTION IN VARIOUS POSITIONS

POSITIONS	EFFECTIVE ϵ = 0.05		EFFECTIVE ϵ = 0.83	
	Per Cent Convection	Per Cent Radiation	Per Cent Convection	Per Cent Radiation
Horizontal heat down	88	12	31	69
45 deg heat down	90	10	38	62
Vertical	93	7	43	57
45 deg heat upward	94	6	48	52
Horizontal heat upward	95	5	55	45

In Fig. 3, the variation of the surface coefficient or surface conductance with temperature difference is shown for various positions and effective emissivities of 0.05 and 0.83. One special test was made in the horizontal position with the heat flowing downward with a baffle, 2 in. high, placed around the calorimeter. This formed a pocket with a consequent lowering of the surface coefficient. The values of the surface coefficients for a temperature difference of 15 F are

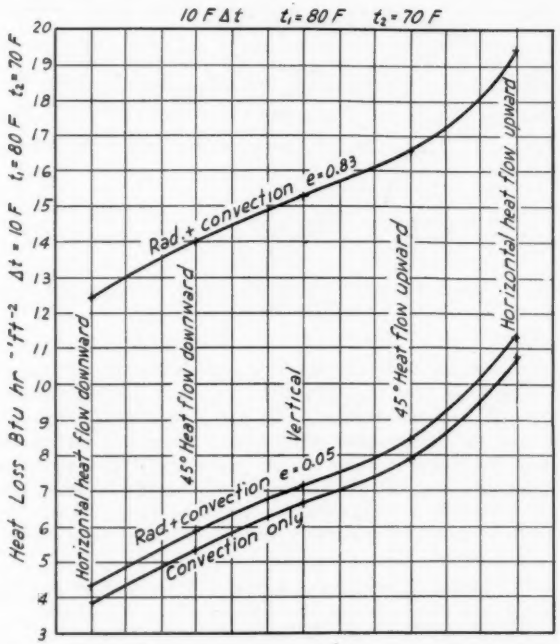

FIG. 2. HEAT LOSS FROM PLANE SURFACE IN VARIOUS POSITIONS

listed in Table 3, and for comparison the values given in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1938.

The values in Table 3 apply only when the surrounding surfaces are at the same temperature as the ambient air. If this is not so, the variation from the generally accepted values might be much greater.

In cases where the surface coefficient is a very important factor in the calculation of heat flow, such as windows and sky lights, concrete roofs, metal roofs and walls, etc., Fig. 4 should be of real use. In this chart, the heat loss by natural convection is separated from that by radiation. The heat loss by convection is given for different positions as well as temperature difference.

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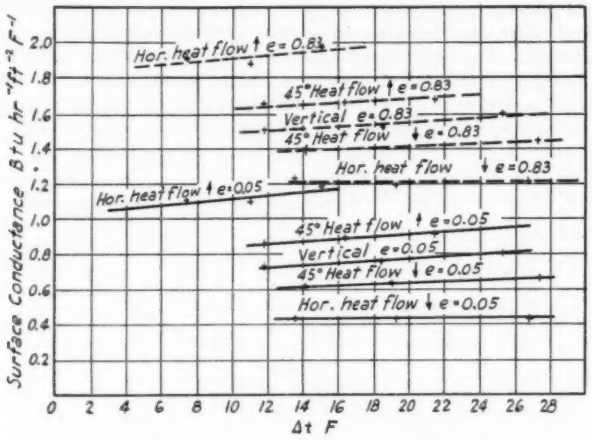

FIG. 3. VARIATION OF SURFACE COEFFICIENT WITH TEMPERATURE DIFFERENCE

Radiation is shown for different effective emissivities as well as temperature difference. The total heat transferred is found by adding the convection to the radiation. If the surrounding walls are at a different temperature than the ambient air, the temperature difference between the surface and the air should be used in finding the convection, while the difference in temperature between the surface and the surrounding walls should be used for radiation. The chart is based on a warm surface temperature of 80 F. For surface temperatures more than a few degrees from 80 F other values should be used.

TABLE 3—SURFACE COEFFICIENTS FOR 15 F TEMPERATURE DIFFERENCE

ORDINARY SURFACES, EFFECTIVE EMISSIVITY 0.83		
Horizontal, heat upward	1.95	GUIDE value 1.65 for all positions and all temperature differences
45 deg, heat upward	1.65	
Vertical	1.52	
45 deg, heat downward	1.39	
Horizontal, heat downward	1.21	
Horizontal, heat downward with special 2 in. baffle	1.13	
REFLECTIVE SURFACE, EFFECTIVE EMISSIVITY 0.05		
Horizontal, heat upward	1.16	GUIDE value 0.80 for all positions and all temperature differences
45 deg, heat upward	0.98	
Vertical	0.74	
45 deg, heat downward	0.62	
Horizontal, heat downward	0.44	
Horizontal, heat downward with special 2 in. baffle	0.35	

For example, the heat loss by radiation from a surface, effective emissivity 0.83, at 60 F and the surrounding surfaces 10 F cooler would be approximately 7.9 Btu per hour per square foot, while the value from the chart based on a surface at 80 F would be 8.7 Btu per hour per square foot.

It has been generally accepted that natural convection varies with approximately the 5/4 power of the temperature difference. This conclusion has been based on a large number of experiments and is approximately true for the

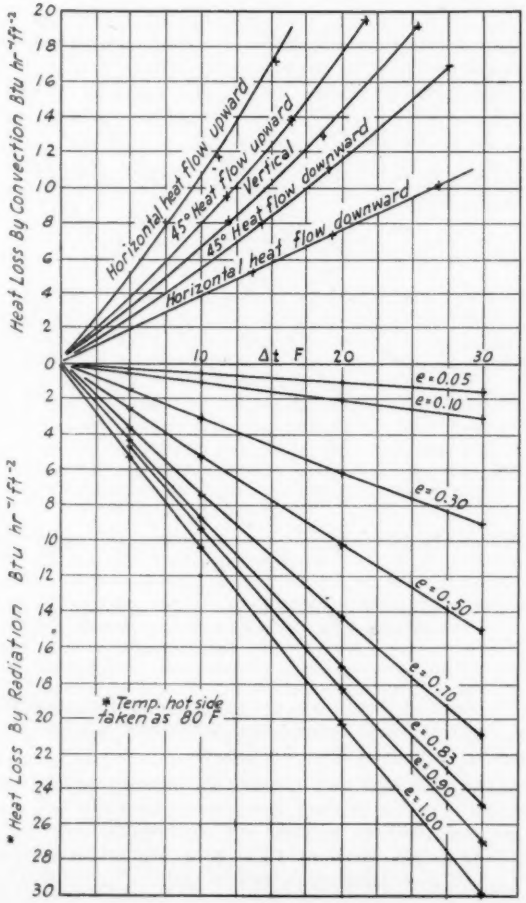

FIG. 4. HEAT LOSS BY CONVECTION AND RADIATION FOR DIFFERENT EFFECTIVE EMISSIVITIES AND TEMPERATURE DIFFERENCES

range of temperatures covered in the experiments. The present tests were made with a much smaller temperature difference than is usual, some being as low as 8 F and the maximum only 28 F. It is interesting to note in Fig. 5, where the logarithm of the convection loss is plotted against the logarithm of the temperature difference that the slope of these lines for various positions is not 5/4 as it should be if the 5/4 law held for these small temperature differences. For four of the lines the slope is 1.12 while the line representing heat flow downward in a horizontal position has a slope of only 1.00. While it is realized that these temperature differences are too small to get very accurate

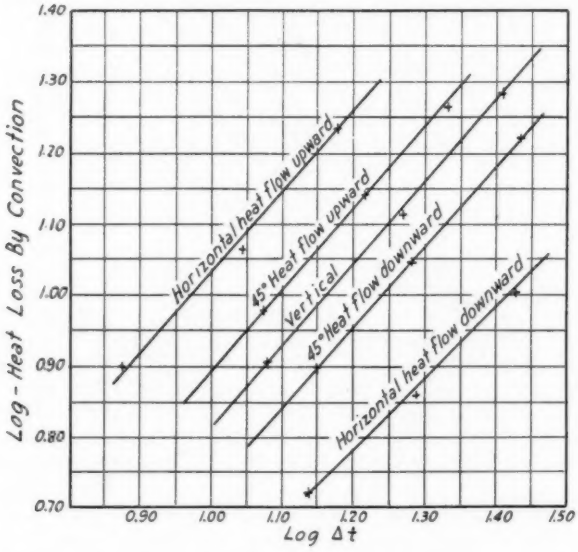

FIG. 5. LOGARITHM OF CONVECTION LOSS PLOTTED AGAINST LOGARITHM OF TEMPERATURE DIFFERENCE

results as to the slope of these lines, the results do indicate that the 5/4 power law may not apply to small temperature differences particularly when in a horizontal position with heat flow downward.

When calculating the heat flow through an ordinary house wall, the inside surface coefficient is not of primary importance and it can be assumed without appreciable error that the air and the surrounding surfaces are at the same temperature. The values given in Table 3 or Fig. 3 for the inside surface coefficient (f_i) will give satisfactory results and the calculation may be made as indicated in THE GUIDE.

It is important to mention, however, that in the case of experimental work for the determination of surface coefficients, it is frequently of more importance

to measure the surrounding surface temperature than that of the air itself. Radiation and convection must be separately evaluated for precise results in this type of work.

There are also cases, in actual buildings, where the inside surface coefficient plays a very important part and where the surrounding surfaces are not at the same temperature as the air. Complicated charts could be made giving f_1 under various conditions of air temperature, surrounding wall temperature, position and emissivity, but it appears less complicated to actually calculate the heat loss under the unusual conditions by a method of approximations.

An extreme case might be represented by radiant wall heating with a wall perhaps 30 F warmer than the air and facing a window that was much cooler than the air. The error could well be more than 25 per cent in such a case if it were calculated by assuming that the air and surrounding surfaces were at the same temperature.

Accurate data on surface coefficients are also of great importance when calculating the possibility of condensation on the inside of windows, skylights, metal roofs, etc.

DISCUSSION

PROF. A. P. KRATZ: In a book on heat transmission[3] is shown a curve in which the logarithm of Nusselt's number is plotted against the logarithm of the product of Grashof's number and Prandtl's number. The equation that he derives from the upper portion of this curve indicates that the heat transfer varies as the 0.25 power of the temperature difference. The slope of the lower portion of the curve, however, indicates that for small temperature differences, or for small values of the product of Grashof's and Prandtl's numbers, the exponent has a value lower than 0.25. The results presented in this paper are apparently consistent with this curve, in that values of the exponent lower than 0.25 were obtained. Have the authors compared their values with the exponent obtained from this curve for a corresponding range? If so, do they find reasonably close agreement?

P. F. MCDERMOTT:[4] The authors are to be congratulated upon the work just reported. Although the heat losses from surfaces to surrounding air have been dealt with by a number of investigators each of whom has made a contribution to our knowledge at least for some specific arrangement, there remains a serious need for more complete and continuous data. We hope that the authors will continue this study to the end of providing such data. Panel sizes should be varied to investigate the influence of dimensions on heat transfer and a range of surface temperatures up to possibly 400 F should be studied. These data should then be correlated with data on cylindrical sections up to several feet in diameter.

Refinements in the data now available are not, commonly, required. However, applications to some problems, for example the insulation of cold pipes or surfaces to prevent condensation, frequently do require more accurate and reliable data than we have.

We are very much interested in the evidence reported that heat transfer by convection is not proportional to the 5/4ths power of the temperature difference. This finding attracts attention to the need for reporting in all conductivity tests on fibrous or porous materials the temperature difference as well as the mean temperature. Many of us have long known that k values frequently vary with the

[3] *Heat Transmission*, by W. H. McAdams, p. 248.
[4] Physicist, Johns-Manville Research Laboratories, Manville, N. J.

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temperature difference more or less independently of the mean temperature for many low density materials. This variation is attributed to the fact that neither convection nor radiation are linear functions of the temperature difference. If the variation in convection is still more complicated, by not even being a constant function of the temperature difference, we will need to exercise more care than ever in dealing with it and in interpreting heat transfer in which it plays a role.

AUTHORS' CLOSURE: "The curve in Professor McAdams' book[6] showing the logarithm of Nusselt's number plotted against the logarithm of the product of Grashof's number and Prandtl's number applies to horizontal cylinders. The lower portion of the curve does indicate that the heat flow for small temperature differences is proportional to approximately the 1.05 power of the temperature difference. This value gradually increases to 1.25 from which point it remains constant for the remainder of the curve. The value of 1.12 found by the authors indicates also that for flat surfaces, the value of the exponent is considerably reduced from 1.25 that is generally accepted for larger temperature differences.

The authors are continuing the work with this apparatus and hope to cover a much larger field. The amount of natural convection in porous and fibrous insulators with relatively small temperature differences is probably not a serious factor. Some tests recently made here indicate that in glass wool of average density, convection is responsible for only about 12 per cent of the total heat flow when the sample is in a horizontal position with the heat flow vertically upward. This would be a maximum condition with less in a vertical position and none in a horizontal position with the heat flow downward.

[6] Loc. Cit. See Note 3.

No. 1099

PERFORMANCE OF SURFACE-COIL DEHUMIDIFIERS FOR COMFORT AIR CONDITIONING

By G. L. TUVE* (*MEMBER*) AND L. J. SEIGEL** (*NON-MEMBER*), CLEVELAND, OHIO

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Case School of Applied Science.

INTRODUCTION AND SUMMARY

EVER since extended surface coils were first introduced in air conditioning, the engineer has been confronted with the problem of finding some method for rating them and for calculating their performance after the ratings have been made. Because of the immediate need for a solution to the problem, a great number of methods have been devised and are now in use. These procedures are based on heat transfer theory to a certain extent, but most of them resort to many approximations, either because the theory does not apply, or because it becomes too complicated for practical use. The purpose of this paper is to review some of the methods available, and to propose another method which combines both simplicity and accuracy. This has been named the ***humidity method.***

In general, the problem may be attacked in two ways: (1) Calculations may be based directly on overall coefficients of heat transfer, U or K. (2) Ratings and calculations may be made on the basis of individual surface coefficients on the refrigerant side and the air side, or on the basis of surface temperatures. If over-all coefficients are used, the ratings are usually prepared in tabular or curve form, and special devices are resorted to for the calculation of dehumidification problems. When surface coefficients are used, the coil can be rated over its entire operating range by means of only a few tests, but the calculations for coil performance are usually more complicated because the methods using surface coefficients are based more exactly on heat transfer theory.

In the first report on this cooperative research project,[1] the two methods of analysis were examined, and the basic theory of dry-coil performance was reviewed. Test results available in the literature were summarized. In the second

* Prof. of Heat Power Engrg., Dept. of Mech. Engrg., Case School of Applied Science.
** Research Asst., Dept. of Mech. Engrg., Case School of Applied Science.

[1] Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation, by G. L. Tuve and C. A. McKeeman. (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 427).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by G. L. Tuve.

TABLE 1—SEVEN METHODS FOR FIGURING DEHUMIDIFYING COILS

Type of Problem: Given, inlet air conditions, total load, heat-load ratio, air velocity, refrigerant velocity, depth of coil.

Required, face area, exit air conditions, refrigerant temperature.

METHOD NO.	TEST RESULTS USED AS BASIS FOR ANALYSIS	PROCEDURE FOR SOLUTION (SEE EQUATIONS, TABLE 3)	ADVANTAGES (See Table 2)	DISADVANTAGES (See Table 2)	REFERENCES (See Footnotes)
1	Over-all coefficients for dry coil as a function of: (1) Air velocity; (2) Depth of coil; (3) Refrigerant velocity.	Assume refrigerant temperature. Solve for exit dry-bulb and for face area by Eqs. A and B, using sensible heat load only. Test answer by experience, and make other trials if necessary.	4, 5, 8	1, 2, 3, 6, 8, 9, 11	ASHVE GUIDE 1937, p. 193
2	Over-all coefficients as in No. 1. Wet-coil multipliers as a function of temperature differences.	Assume refrigerant temperature. Solve for exit dry-bulb and for face area by Eqs. A and B, using wet-coil coefficient and total heat load. Make other trials if necessary.	4, 8	1, 2, 5, 7, 8, 9, 10	Commercial Bulletins
3	Over-all coefficients as in No. 1. Wet-coil multipliers as a function of heat-load ratio. Exit relative humidities as a function of heat-load ratio.	Locate exit relative humidity on load ratio line. This fixes exit conditions. Solve for face area by Eq. B and for refrigerant temperature by Eq. A, using wet-coil coefficient and total load.	1, 3, 4, 6, 8	2, 5, 7, 8, 10	(3)
4	Over-all coefficients for sensible and latent heat transfer, given separately. (Latent heat coefficients based on dew-point mean temperature difference.)	Assume refrigerant temperature. Solve for exit dry-bulb and exit dew-point by Eq. A, using U_S and U_L. If exit condition does not fall on load ratio line, make other trials.	3, 4, 5, 6, 8	1, 2, 7, 8, 10	(3)
5	Ratio of heat removed to heat removable, as a function of air velocity and depth of coil. Refrigerant - to - surface temperature differential as a function of total load, ($\triangle T$).	Using intersection of load ratio line with saturation curve as surface temperature, find exit conditions by ratio of heat removed to heat removable. Solve Eq. B for face area. Find refrigerant temperature by subtracting $\triangle T$ from surface temperature.	1, 3, 4, 6, 8	4, 5, 6, 8, 10	(6), (7)
6	Surface coefficients as a function of air and refrigerant velocities.	Assume refrigerant temperature. Find extent of dry surface, if any. Solve for final wet-bulb, and locate intersection on load ratio line. (Tables of empirical factors required.)	1, 2, 3	4, 5, 8, 10	(4)

TABLE 1—(*Continued*)

7	Surface coefficients as a function of air and refrigerant velocities.	Find minimum dry-bulb for zero dehumidification by Eq. F, with T_S = entering dew-point. Draw a relative humidity line through this dry-bulb and entering dew-point. Find face area from Eq. B, surface temperature from Eq. F and refrigerant temperature from Eq. D.	1, 2, 3, 4, 5, 6, 7, 8	8, 10	This Paper

report,[2] additional experimental data on dry coils were presented, and a method was suggested as a basis for dry-coil rating and calculation, using overall coefficients. The third report [3] dealt with dehumidification, and showed the approximate variation of performance in the range of commercial coil designs. Here again the possibilities of several methods based on overall coefficients were examined. The present paper deals with the application of the surface coefficient method for dehumidifying coils. A simple method of solution on the psychrometric chart is given, and its applications to coil rating and coil selection are discussed. Solution by trial-and-error becomes unnecessary with this *humidity method,* and coil performance over the entire range can be determined

TABLE 2—ADVANTAGES AND DISADVANTAGES OF VARIOUS METHODS OF FIGURING DEHUMIDIFYING COILS

(*See Table* 1)

ADVANTAGES (Characteristics of the ideal method)

1. Trial-and-error solution not required.
2. Few tests necessary for rating a coil.
3. Good accuracy for coils of the type tested.
4. Method easily explained from psychrometric chart and simple heat transfer equations.
5. Tables of empirical factors not necessary.
6. Easy graphical solution on psychrometric chart.
7. Wet-coil tests not required for establishing ratings.
8. Calculations very simple.

DISADVANTAGES (of actual methods)

1. Many problems require trial-and-error solutions.
2. Large number of tests necessary for rating coils.
3. Uncertain accuracy, probably not closer than 10 per cent.
4. Method difficult to understand in terms of fundamentals.
5. Extensive tables of constants or empirical factors required.
6. Inaccurate for high latent heat loads.
7. Difficult tests required for establishing ratings.
8. Applicable only to specific coils tested.
9. Applicable only to operating conditions used in tests.
10. More research necessary to establish accuracy.
11. Coil may not dehumidify according to specified heat-load ratio.

[2] Performance of Fin-Tube Units for Air Heating, Cooling and Dehumidifying, by G. L. Tuve. (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 99).

[3] Performance of Fin-Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 357).

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from the results of a few simple dry-coil tests. The method is applicable to both direct-expansion and water coils.

One of the inherent advantages of surface-coil dehumidifiers over the spray type is the higher exit dry-bulb which is possible with a given amount of dehumidification. This may result in a smaller required size of refrigerating machine, and the necessity for little or no reheating. As soon as the performance characteristics of surface coils are better understood by air conditioning engineers, these advantages will be more widely recognized.

CORRELATIONS BETWEEN DRY-COIL AND WET-COIL PERFORMANCE

A practical summary for comparing most of the methods now in use for figuring dehumidifying coils is given in Table 1. A single type of problem is used as a basis for this comparison, *i.e.*, the problem of finding the exit air conditions, the coil size and the required temperature of refrigerant when the inlet air conditions and the heat load ratio are known. In methods 1 to 4, in Table 1, the over-all coefficient of heat transfer (U or K) is used. Methods 6 and 7 utilize the individual air-side and refrigerant-side coefficients. Methods 5 to 7 also employ some form of surface temperature in the calculation.

Reference is made in Table 1 to the advantages and disadvantages of the various methods, as indicated by the list in Table 2. An ideal method for rating and calculating dehumidifying coils would have all of the favorable characteristics listed in the first part of Table 2 and none of the disadvantages enumerated in the last part of that table. From Table 1 it is evident that, while the early methods of figuring surface dehumidification had many advantages, they also had certain important disadvantages, chief of which were their lack of general application and their uncertain accuracy.

The proposed *humidity method* of coil rating and calculation (No. 7, in Table 1), has most of the advantages and few of the disadvantages listed in Table 2. This method deserves further investigation, as outlined in the last section following.

The procedure outlined in Table 1 for each of the seven methods is not in all cases the only procedure available. For instance, in method No. 2, an assumed wet-bulb depression at the exit, or relative humidity of the exit air may be used as the starting point, rather than an assumed refrigerant temperature. Again, in using method No. 5, the refrigerant-to-surface temperature differential is sometimes found by an arbitrary rule or approximation, rather than from tests.

There are also several types of problems other than the one used as a basis for Table 1 and the procedure for solving some of these is illustrated in Table 4. Trial-and-error solutions are necessary for many types of problems when methods 1 to 6 are used, while method No. 7 usually affords a direct solution, without trial-and-error.

SURFACE TEMPERATURES AND SURFACE COEFFICIENTS

Many of the solutions to surface-coil problems now in use have been developed through a knowledge of some form of surface temperature. The true surface temperature of a coil is a variable, depending on many factors in

the coil's construction and operation. Goodman has shown that surface temperature varies in the direction of the air flow,[4] but equally important are the variations due to the temperature rise of the refrigerant and to the temperature gradient in the fins. Some investigators have attempted to determine surface temperatures by direct measurement with thermocouples or by theoretical calculation from other measured quantities.[5] To take the place of true surface temperature, fictitious surface temperatures have been worked out and used by different experimenters. For example, the intersection of the heat-load

TABLE 3—EQUATIONS

(*See List of Symbols, Table* 8)

Eq. (A): Total heat transfer = over-all coefficient × total surface area × log mean temperature difference.

$Q_T = U \times (A \times \text{depth of coil} \times \text{face area}) \times mtd_1$

or,

$Q_T = h \times (A \times \text{depth of coil} \times \text{face area}) \times mtd_2$

where mtd_1 is measured from fluid to fluid, and mtd_2 is measured from fluid to surface.

Eq. (B): Sensible heat transfer = weight of air × specific heat × temperature change.

$Q_S = G \times \text{face area} \times 0.24 \times (T_1 - T_2)$

$Q_S = 0.24 \times \text{cfm} \times 60 \times 0.075 \times (T_1 - T_2)$

$Q_S = 0.24 \times \text{face velocity} \times \text{face area} \times 60 \times 0.075 \times (T_1 - T_2)$

Eq. (C): Total heat transfer = weight of air × change in enthalpy.

$Q_T = \text{G} \times \text{face area} \times (H_1 - H_2)$

$Q_T = \text{face velocity} \times \text{face area} \times 60 \times 0.075 \times (H_1 - H_2)$

Eq. (D): $Q_T = h_r/R\ (T_S - T_r)\ AN$

or,

$$T_S = T_r + \frac{RQ_T}{h_r AN}$$

Eq. (E): $1/U = 1/h_a + R/h_r$ ..... (Appendix 3)

Eq. (F): $h_a = (0.24G/AN) \log_e \dfrac{T_1 - T_S}{T_2 - T_S}$ ..... (Appendix 2)

ratio line with the saturation line on the psychrometric chart has been given the interpretation of surface temperature by Knaus,[6] Wells,[7] Pownall [8] and others. (Table 1, method No. 5.) Usually a *mean* surface temperature is used, which is supposed to take care of the temperature gradient in the fins.

In this paper, surface temperature has been considered as the temperature of an imaginary surface having the same characteristics for heat transfer as the coil itself. To obtain a clearer idea of this interpretation, assume that

[4] Dehumidification of Air with Coils, by William Goodman. (*Refrigerating Engineering*, October, 1936, p. 225).

[5] Extended Surface Cooling Units, by R. H. Swart. (*Refrigerating Engineering*, February, 1938, p. 107).

[6] Heat Transmission in Cooling Air with Extended Surfaces, by W. L. Knaus. (*Refrigerating Engineering*, January, 1935, p. 23, February, 1935, p. 82).

[7] Graphical Method of Determining Finned Coil Capacities Described by E. P. Wells. (*Heating, Piping and Air Conditioning*, December, 1936, p. 665).

[8] Rational Development and Rating of Extended Air Cooling Surface, by H. B. Pownall. (*Refrigerating Engineering*, October, 1935, p. 211).

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the heat transfer coefficients have been determined for the coil under consideration. Then imagine the whole coil to be a thin-walled pipe or bundle of tubes having the same heat transfer coefficients as the coil, but consisting of prime surface only. Consider a refrigerant to be inside this pipe, and air to be blowing across the pipe on the outside. It is the average temperature of this type of imaginary surface that is called surface temperature, T_s in the development which follows.

Surface coefficients are usually determined from over-all coefficients by some direct or indirect use of surface temperature in heat transfer equations, as in the method developed by Goodman.[9] However they are determined they may

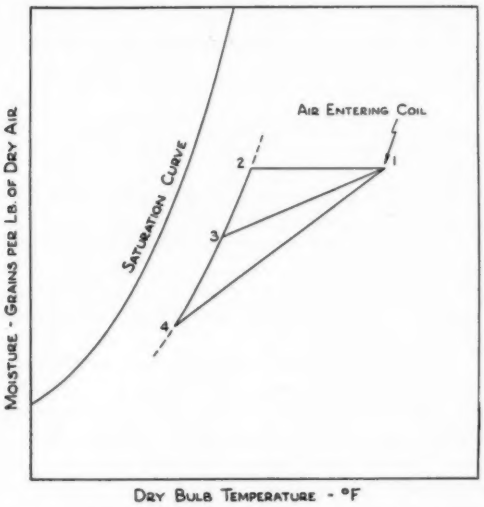

FIG. 1. SKETCH OF HUMIDITY METHOD ON PSYCHROMETRIC CHART

be expressed in terms of the velocity of the fluid with which they are associated, and they may be used to calculate the over-all coefficients for any combination of conditions, by equation E, (Table 3). When calculations of the over-all coefficient are made for a dehumidifying coil, due account must be taken of the latent heat as well as the sensible heat transfer.

The refrigerant-side surface coefficient is independent of what happens outside the coil; hence it is no doubt identical for both dry-cooling and dehumidification. The temperature gradient (or temperature rise) across the inside surface film is directly proportional to the total load; hence it changes when dehumidification begins, as does also the temperature gradient in the fins. The outside surface coefficient and the temperature gradient across the air film are both changed by the presence of dehumidification. Methods for taking

[9] Loc. Cit. See Note 4.

these changes into account, and for determining the surface coefficients are given in the following section.

DERIVATION AND USE OF THE PROPOSED HUMIDITY METHOD

Before indicating the derivation of method No. 7, Table 1, it is well to review the characteristics of dehumidifying coils as indicated by the heat-load ratio lines on a psychrometric chart. If the inlet and exit air conditions are located on a non-logarithmic psychrometric chart (for example, points 1 and 4, Fig. 1) the angle which the straight line connecting the two points makes with the horizontal is a direct measure of the relative amounts of sensible heat and of latent heat extracted from the air by the coil, *i.e.*, this angle is a measure of the heat-load ratio. Conversely, if the initial condition of the air is located on the chart, and the required heat-load ratio line drawn through it, the exit condition will be somewhere on this line, near the saturation curve. In this latter case, the problem is to locate the exit point, usually by means of another line which will intersect the load-ratio line. The exit air point may thus be located as follows:

1. By finding the dry-bulb temperature of the exit air (Method No. 1, Table 1).
2. By finding the wet-bulb temperature of the exit air.
3. By finding the dew-point temperature of the exit air (Method No. 4).
4. By finding the relative humidity of the exit air.
5. By finding the wet-bulb depression (below dry-bulb) for the exit air.
6. By finding the ratio: Heat removed/heat removable = $\frac{\text{Length of line from inlet to exit air conditions}}{\text{Length of line from inlet to saturation curve}}$ (Method No. 5)

The method now to be described is approximately equivalent to finding the relative humidity of the exit air.

Suppose air enters a cooling coil at conditions corresponding to point 1 in Fig. 1. As long as the surface temperature of the coil is above the dew-point, the air will be cooled without loss of moisture, and its condition as it leaves the coil will be somewhere along the line 1-2. Its exact position on this line will depend on the air velocity and the air-side surface coefficient. When the surface temperature just equals the dew-point, the air will leave with conditions represented by point 2. But if the surface temperature is reduced below the dew-point, condensation will take place, and *the air will have a final condition somewhere along the line 2-3-4, which is a curve at a constant horizontal distance from the saturation curve. It must be understood that the path 1, 2, 3, 4, is not intended to represent the path of the condition of the air as it passes through the coil, but rather the path traced by the exit air conditions as the surface temperature is gradually reduced, with other conditions remaining constant.* For proof of the statement that the exit conditions lie along this line, see the section on Experimental Work, and also Appendix 1.

The horizontal distance from the saturation curve to a given relative humidity line on the psychrometric chart is very nearly constant.[10] Therefore the minimum dry-bulb temperature for zero dehumidification (represented by point 2 in Fig. 1) and the exit conditions for any heat-load ratio, should have ap-

[10] See Relation of Dew-Point to Relative Humidity. (HEATING, VENTILATING, AIR CONDITIONING GUIDE 1938, Chapter 1, p. 9).

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proximately the same relative humidity. Hence, method No. 7, Table 1, has been named the relative humidity method, or merely the *humidity method.*

Summary of Procedure for Humidity Method

1. Solve Eq. F, Table 3 for the minimum exit dry-bulb temperature which can be attained without dehumidification, and establish the intersection on the psychrometric chart of this temperature and the horizontal through the entering dew-point.
2. Through the point just established, draw a line of constant relative humidity. The intersection of this line with *any* load-ratio line determines the exit air conditions for that heat-load ratio.
3. Solve Eq. B, Table 3, for face area.
4. Find surface temperature by Eq. F.

There are several other types of problems which are important in fin-coil rating or selection. Most of these are included in the summary of Table 4, which indicates the application of the *humidity method* to five other groupings of the knowns and unknowns besides the one used for Table 1. Trial-and-error solutions are not necessary in any of these except No. 1.

The proposed method has certain limitations with respect to the refrigerant conditions. Since a single surface temperature is used, the accuracy of the method is uncertain when the temperature rise of the water is large, as it may be in deep counterflow coils. With a direct-expansion refrigerant, the method of finding surface coefficients outlined in Appendix 4, is not suitable. One way to overcome this difficulty would be to test a similar coil with water refrigerant, and then use the same air-side surface coefficients for the direct-expansion coil.

A special advantage of the *humidity method* is that it gives a solution for cases in which the latent heat load is high. In those methods which use the intersection of the heat-load ratio line with the saturation curve as the surface temperature (see method No. 5, Table 1), it may be impossible to locate this intersection when the latent heat load is great.

EXPERIMENTAL WORK

In order to examine the validity of the method just described, and of the derivations given in the Appendix, three fin-coils of radically different designs were tested.

The test equipment and the methods used were similar to those described in a previous report.[11] A line diagram of the test unit is shown in Fig. 2, and Fig. 3 is a photograph of the set-up.

The equations obtained for surface coefficients by analysis of the tests on the three designs of coils are given in Table 5. These were dry-coil tests.

The method for finding the surface coefficients was suggested by McAdams,[12] and has been used by Pownall[13] and others. The procedure is outlined in Appendix 3. Air-side coefficients are expressed as a function of air velocity, and water-side coefficients as a function of water velocity.

In addition to the dry-coil tests for determining surface coefficients, many dehumidifying tests were made. For each test condition a comparison was made

[11] Loc. Cit. See Note 3.
[12] Heat Transmission, by W. H. McAdams, p. 264.
[13] Loc. Cit. See Note 8.

TABLE 4—PROCEDURE FOR SOLVING PROBLEMS BY METHOD NO. 7[a]

	PROBLEM		PROCEDURE FOR SOLUTION
	GIVEN	REQUIRED	
1	Ent. Cond. Air velocity. Refrigerant velocity. Refrigerant temperature.	Exit Cond. Load ratio. Total heat transfer.	1. Find minimum dry-bulb for zero dehumidification by Eq. F with T_S = Entering *DP*. 2. Find T_S from Eq. D. 3. Find exit DB from equation F. 4. Draw a relative humidity line through the entering *DP* and DB found in (1). At the intersection with the DB found in (3), read exit WB. 5. Find total heat transfer from Eq. C. 6. Find sensible heat transfer and load ratio.
2	Ent. Cond. Air velocity. Refrigerant velocity. Load ratio.	Exit Cond. Refrigerant temperature. Total heat transfer.	1. Find minimum dry-bulb for zero dehumidification by Eq. F with T_S = Entering *DP*. 2. Draw a relative humidity line through the entering *DP* and DB found in (1). At the intersection with the load ratio line, read exit conditions. 3. Find total heat transfer from Eq. C. 4. Find T_S by Eq. F and T_r by Eq. D.
3	Ent. Cond. Air velocity. Load ratio. Refrigerant temperature.	Exit Cond. Refrigerant velocity. Total heat transfer.	1. Find minimum dry-bulb for zero dehumidification by Eq. F with T_S = Entering *DP*. 2. Draw a relative humidity line through the entering *DP* and DB found in (1). At the intersection with the load ratio line, read exit conditions. 3. Find total heat transfer from Eq. C. 4. Find T_S by Eq. F. 5. Find h_r/R by Eq. D and calculate V_r.
4	Ent. Cond. Refrigerant velocity. Total heat transfer Exit Cond.	Load ratio. Refrigerant temperature. Air velocity.	1. Find air velocity by Eq. C. 2. Calculate sensible heat transfer and load ratio. 3. Find T_S by Eq. F. 4. Find T_r by Eq. D.
5	Ent. Cond. Exit Cond. Load ratio. Total load. Air velocity. Face area. Refrigerant velocity (or surface coefficient).	Coil depth. Refrigerant temperature.	1. Draw a relative humidity line through the exit conditions, and at the intersection of this line with a straight line through the entering *DP*, read the minimum DB for zero dehumidification. 2. Calculate h_a from air velocity and solve Eq. F for number of rows of coil depth, (N). 3. Solve Eq. F for T_S. 4. Find T_r from Eq. D.

[a] Problems 1 to 4 show how to find the capacity of a given coil. Problem 5 shows how to select a coil size for a given service.

between the calculated relative humidity of the exit air for maximum cooling without dehumidification, and the measured relative humidity at exit during the actual test. In other words, the relative humidities at points 2 and 3, Fig. 1, were compared. These comparisons are shown in the last two columns of Table 6. It will be noted that the agreement is very good in spite of a wide variety

of operating conditions. While three different makes of coils are represented in Table 6, they were all 4-row cross-flow coils, and in order to examine the results with other coil depths, a number of the tests reported by Goodman [14] were analyzed in a similar manner (see Table 7). The validity of both Goodman's method and the *humidity method* is indicated by the fact that those test points which fell on his curves also showed a good check by the *humidity method*, but the points which departed from his curves failed to give an agreement between calculated humidity and test results.

LIMITATIONS OF ALL COIL-RATING METHODS

There is some question whether the actual performance of a dehumidifying coil under seemingly identical conditions is consistent within ± 5 per cent, and

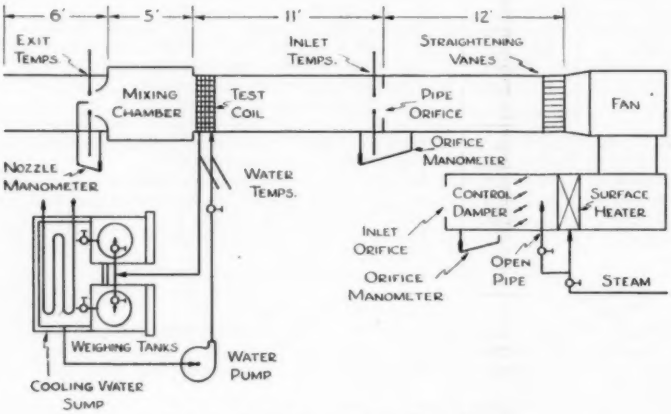

FIG. 2. LINE DIAGRAM OF TEST UNIT

therefore whether *any* method can predict coil performance much closer than this. It has been noted before that a wet-coil test may be repeated under identical conditions of entering air temperatures and velocity, identical entering refrigerant temperature and velocity, and identical test set-up, and the coil capacity may vary 5 per cent, even when the heat-balance checks closer than 2 per cent.

It was suggested in a previous paper,[15] that these variations are due to differences between the latent heat coefficients, depending on the predominance of drop-condensation or film-condensation. Schmidt and others [16] have reported differences greater than 400 per cent in coefficients of heat transfer for steam condensation on surface-condenser tubes with identical operating conditions but different previous history of the tube-surface.

[14] Loc. Cit. See Note 4.
[15] Loc. Cit. See Note 3.
[16] Loc. Cit. See Note 12, p. 257.

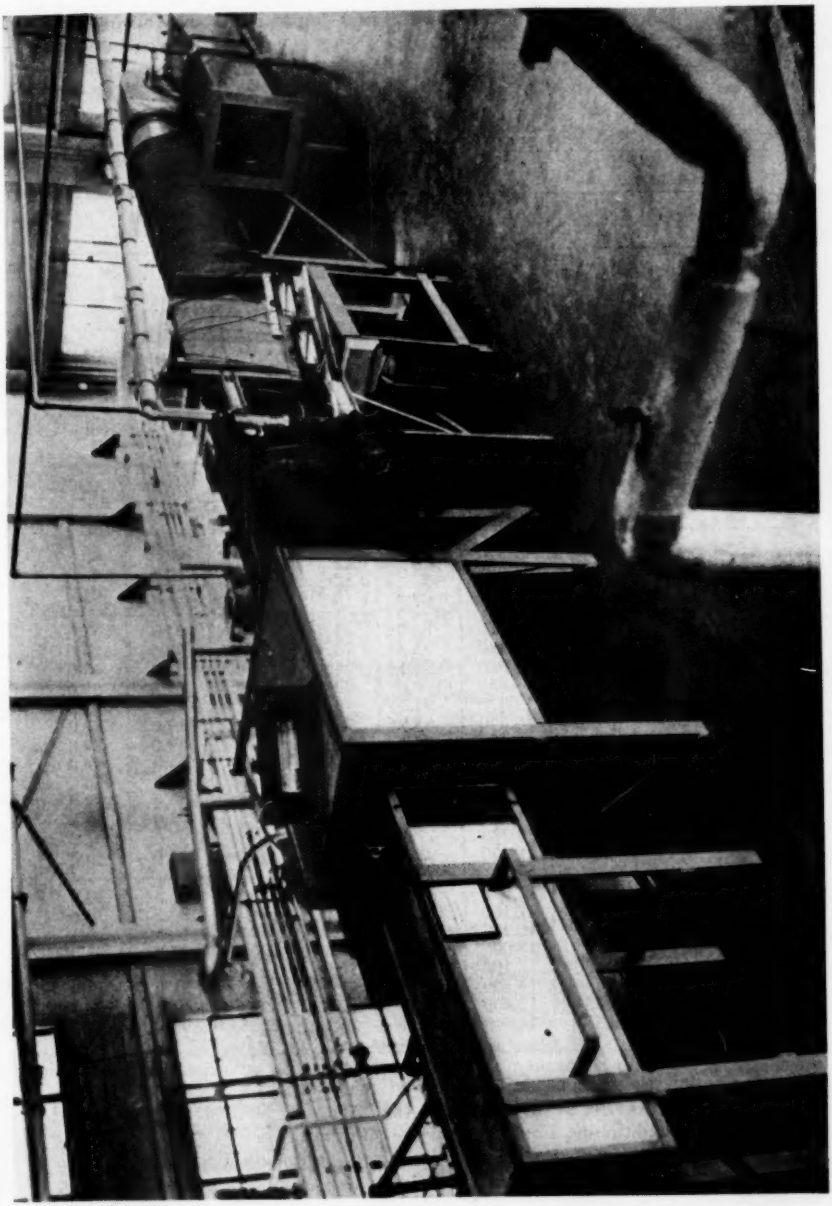

FIG. 3. PHOTOGRAPH OF TEST SET-UP

Another difficulty in all coil tests is the near impossibility of securing sufficiently thorough mixing of the exit air so that temperatures can be measured to one or two tenths of a degree. The difficulty is especially great with wet-bulb readings. The type of mixing-box specified in the Standard Code for

TABLE 5—SURFACE COEFFICIENTS

COIL	SURFACE COEFFICIENT	
	AIR-SIDE (h_a)	REFRIGERANT-SIDE (h_r/R) REFERRED TO AIR-SIDE AREA
A	$0.0588\ V_a^{0.88}$	$17.55\ V_r^{0.8}$
B	$0.405\ V_a^{0.47}$	$22.7\ V_r^{0.8}$
C	$0.130\ V_a^{0.76}$	$21.4\ V_r^{0.8}$

TABLE 6—COMPARISON OF TEST RESULTS WITH COMPUTED RESULTS

COIL	TEST NO.	LOAD RATIO $\frac{S}{T}$	TOTAL LOAD BTU/HOUR SQ FT FACE	ENTERING CONDITIONS		AIR VEL. FPM	RELATIVE HUMIDITY	
				Dry-bulb	Wet-bulb		At Exit (By Test)	At Td (Computed)
A	15	0.638	11,320	83.9	74.0	477	90.0	90.5
	16	0.444	13,000	86.0	78.5	324	92.0	93.0
	17	0.534	13,830	86.2	76.6	415	91.0	91.0
B	5	0.516	9,350	83.0	75.0	273	93.0	94.0
	6	0.302	9,430	75.3	75.3	271	99.5	100.0
	12	0.634	10,050	84.1	74.4	540	92.0	93.0
	13	0.895	15,860	102.0	82.5	685	83.0	83.0
	32	0.638	13,920	91.7	79.5	578	90.5	90.5
	33	0.802	10,090	83.9	72.0	574	89.0	89.0
	34	0.795	11,600	85.4	72.6	571	89.0	89.0
	36	0.703	9,870	84.5	74.1	463	91.0	91.0
	37	0.809	8,620	85.4	74.0	474	90.0	90.0
	38	0.894	9,610	81.3	70.1	635	88.0	88.0
	39	0.596	14,100	83.5	75.5	635	92.0	92.0
	40	0.652	9,280	77.1	70.5	632	93.0	93.0
	41	0.608	11,200	91.8	77.8	274	93.0	93.0
C	29	0.670	12,920	88.8	73.1	389	79.5	80.0
	10	0.730	17,160	100.0	78.0	363	80.5	79.5
	15	0.440	16,300	85.7	77.9	391	89.5	90.5
	11	0.650	18,000	100.0	80.1	369	82.0	82.0
	30	0.640	17,100	89.8	76.0	483	84.0	85.0
	8	0.460	19,500	89.9	79.8	399	87.0	89.0
	25	0.735	12,400	90.0	72.4	295	83.0	83.0
	6	0.350	14,000	79.8	76.1	385	95.0	96.0

Testing and Rating Steam Unit Heaters of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Standard Method of Rating and Testing Air Conditioning Equipment of the *American Society of Refrigerating Engineers* is very unsatisfactory in this regard, and suggestions for improvement are solicited.

Still another limitation on the accuracy of methods of rating dehumidifying coils is the presence of the water of dehumidification. Many designs of coils do not drain steadily, or the exact nature of the condensate film depends on just how the coil is installed, upon its cleanliness, and upon its immediate past history. All of these limitations call for additional attention to the problem of the inherent consistency and reproducibility of wet-coil operation, and hence the probable limits of accuracy of coil ratings.

TABLE 7—EXPERIMENTAL RESULTS BY GOODMAN RECALCULATED BY HUMIDITY METHOD.[a]

(*See Reference No.* 4)

TEST NO.	NO. OF ROWS	LOAD RATIO	ENTERING CONDITIONS			RELATIVE HUMIDITY	
			Dry-bulb	Wet-bulb	Air Velocity Fpm	By Test Per Cent	Calculated Per Cent
The following points fall close to Goodman's condition curves:							
16	4	0.795	84.0	68.4	654	80.5	80.5
16	5	0.775	84.0	68.4	654	87.0	87.0
16	6	0.760	84.0	68.4	654	89.0	89.5
16	7	0.755	84.0	68.4	654	93.0	93.0
17	5	0.500	85.9	76.1	694	92.0	92.0
17	6	0.480	85.9	76.1	694	91.0	93.0
18	5	0.600	87.4	74.2	689	88.0	88.0
18	6	0.580	87.4	74.2	689	89.0	91.0
25	5	0.900	106.1	75.1	702	75.0	75.0
These points *do not* fall close to Goodman's condition curves:							
17	7	0.462	85.9	76.1	694	89	95
18	7	0.560	87.4	74.2	689	90	94
25	7	0.825	106.1	75.1	702	80	85

[a] These data were based on Table 7 in Goodman's paper.

SUGGESTED PROGRAM FOR FIN-TUBE RESEARCH

In order to develop an acceptable uniform method for rating and comparing dehumidifying coils and for calculating their performance under a variety of conditions, the following program for the continuance of this research should be carried out as rapidly as possible:

1. *Variation of air-side surface coefficients.* The variation of the air-side coefficient should be studied, especially with reference to the comparison of dry- and wet-coils, and at various heat-load ratios. These determinations will furnish a further check on the accuracy of the *humidity method* for obtaining coil performance.
2. *Refrigerant-side coefficients for water.* The effect of water velocities on the water-side coefficients should be further studied, especially for the higher water velocities now becoming common. The actual advantages of the counter-flow arrangement as compared with cross-flow should be demonstrated by comparative tests.

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3. *Refrigerant-side coefficients for direct expansion.* For dichlorodifluoromethane in particular, a study of refrigerant-side coefficients for approved methods of coil arrangement will clear up some of the uncertainties which exist in the comparisons between direct-expansion and water coils. It will also give definite information regarding the amount of dehumidification possible at various low-side pressures (or refrigerant temperatures). Suitable experimental methods for dichlorodifluoromethane-coil tests should be developed and specified.

4. *Depth of coils.* The effect of the depth of coil upon its heat transfer characteristics should be studied further, such as by tests on coils 2, 4, 6, 8, and 10 rows deep.

5. *Relation of heat transfer to air velocity and friction.* Further studies should be made of the economics of coil design and selection with respect to heat transfer and air friction.

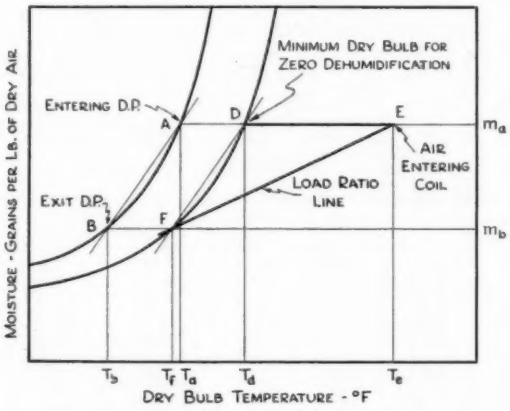

FIG. 4. DEVELOPMENT OF HUMIDITY METHOD

The suggested program should cover both continuous-fin and round-fin coils. All work should be directed toward the setting up of uniform Test Code methods for rating and calculating surface coils. The accuracy of the various methods should be studied further.

APPENDIX NO. 1

Proving that: *The locus of the exit air conditions from a coil, as refrigerant temperature is varied, is a line on the psychrometric chart at a constant horizontal distance from the saturation curve.*

The proof of the method proposed for analysis of wet-coil performance will now be given. For this purpose it will be necessary to derive an expression for heat-load ratio in terms of entering air conditions and refrigerant temperature.

From the psychrometric chart, the moisture content corresponding to any dew-point may be found in terms of temperature if the equation of the saturation curve is known in terms of moisture content and temperature. However, a simpler method for finding moisture content in terms of temperature can be used in this derivation.

Referring to Fig. 4, if a straight line AB is drawn through the points on the saturation curve which represent the entering and final dew-points of air being cooled by a dehumidifying coil, the values of the moisture content at these dew-points may be expressed in terms of the slope and intercept of that line. The general expression is: (See List of Symbols, Table 8.)

$$m = dm/dT\ (T) + C \quad \ldots\ldots (1)$$

Therefore, the moisture difference between the entering and final conditions may be represented by the following expression which was obtained directly from equation (1) by solving for m_a and m_b and then subtracting the results.

$$m_a - m_b = dm/dT\ (T_a - T_b) \quad \ldots\ldots (2)$$

where the subscripts a and b refer to any dew-points whatever. The slope dm/dT depends on the actual values of T_a and T_b, that is, the slope of the straight line connecting the initial and final dew-points varies with the position of those dew-points on the saturation curve.

The line EF in Fig. 4 is the load-ratio line corresponding to any coil performance in which the air enters the coil with the moisture content m_a, and leaves the coil with the moisture content m_b. The latent heat extraction through the coil is:

$$Q_L = \frac{L}{7000}\ (m_a - m_b)$$

$$= \frac{L}{7000}\ (m_a - m_S)\ f\ (V_a) \quad \ldots\ldots (3)$$

where $(m_a - m_S)$ equals all the moisture which would be abstracted from the air if it were cooled down to the coil surface temperature, and $f\ (V_a)$ takes care of the fact that it is cooled only a portion of this distance. In other words, the ratio of the latent heat removed to the maximum removable with the given surface temperature is a function of the air velocity (and, of course, of the coil design and coil depth).

Replacing $(m_a - m_b)$ in Equation (3) by its equivalent from Equation (2), the latent heat transfer may be expressed in terms of temperatures rather than moisture contents:

$$Q_L = \frac{L}{7000}\ \frac{dm}{dt}\ (DP - T_S)\ f\ (V_a) \quad \ldots\ldots (4)$$

TABLE 8—LIST OF SYMBOLS

dm/dT =	Slope of straight line connecting initial and final dew-points (on the saturation curve).
m =	Moisture content in grains per pound of dry air.
$m_a = m_2$ =	Moisture content at A in grains per pound of dry air.
$m_b = m_3$ =	Moisture content at B in grains per pound of dry air.
T =	Dry-bulb temperature.
T_a =	Dry-bulb temperature (and dew-point) at A.
T_b =	Dry-bulb temperature (and dew-point) at B.
$T_1 = T_e$ =	Entering dry-bulb temperature.
$T_2 = T_d$ =	Minimum dry-bulb temperature for zero dehumidification.
$T_3 = T_f$ =	Exit dry-bulb temperature.
T_r =	Refrigerant temperature.
T_S =	Surface temperature.
Q_S =	Sensible heat transfer.
Q_L =	Latent heat transfer.
Q_T =	Total heat transfer.
DP =	Entering dew-point.
L =	Latent heat of vaporization.
K =	$L/0.24 \times 7000$.
G =	Weight of air in pounds per hour per square foot of face area.
h_a =	Air-side surface coefficient in Btu per hour per square foot surface per degree, sensible heat.
V_a =	Face velocity, feet per minute.
h_r =	Refrigerant side surface coefficient in Btu per hour per square foot surface per degree.
V_r =	Refrigerant velocity, feet per second.
R =	Ratio of air-side to refrigerant-side surface.
A =	Air-side surface area in square feet per square foot of face per row coil depth.
U =	Overall coefficient in Btu per hour per square foot surface per degree.
N =	Number of rows of coil depth.

It has been assumed by several investigators[17] that the ratio of heat removed to heat removable is similarly affected by air velocity, whether latent heat, sensible heat or total heat is being considered. Making this assumption, the following may be written:

$$Q_S = 0.24\,(T_1 - T_S)\,f\,(V_a)$$

But:

$$Q_L + Q_S = Q_T;\ \text{or}\ Q_T/Q_S = Q_L/Q_S + 1$$

Therefore:

$$Q_T/Q_S = \frac{L\ dm/dt\ (DP - T_S)\,f\,(V_a)}{0.24 \times 7000\ (T_1 - T_S)\,f\,(V_a)} + 1 = \frac{K\ dm/dt\ (DP - T_S)}{(T_1 - T_S)} + 1 \qquad (5)$$

From which:

$$T_S = \frac{K\ dm/dt\ DP - T_1\,(Q_L/Q_S)}{K\ dm/dt - Q_L/Q_S} \qquad (6)$$

Now, select any set of entering conditions and hold them constant (as well as the air velocity and refrigerant velocity) and let the surface temperature be varied by variation of refrigerant temperature. Under these conditions, decreasing surface temperatures will result in increasing latent heat transfer, and the result on a psychrometric chart will be similar to Fig. 1: where line 2, 3, 4 is a curve at a constant horizontal distance from the saturation curve. This statement will now be proved by substituting different values of surface temperature in Equation F. (See appendix No. 2.)

$$\frac{h_a A N}{0.24\ G} = \log_e \frac{T_1 - T_{S1}}{T_2 - T_{S1}} = \log_e \frac{T_1 - T_{S2}}{T_3 - T_{S2}}$$

The assumption that h_a for sensible cooling is the same for both wet-and dry-surface has been made by other investigators[18] and is required here.

Then,

$$\frac{T_1 - T_{S1}}{T_2 - T_{S1}} = \frac{T_1 - T_{S2}}{T_3 - T_{S2}} \qquad (7)$$

Substituting (6) in (7) and letting $X = K dm/dT$ and $Z = Q_L/Q_S$

$$\frac{T_1 - \dfrac{X\ DP - Z_1T_1}{X - Z_1}}{T_2 - \dfrac{X\ DP - Z_1T_1}{X - Z_1}} = \frac{T_1 - \dfrac{X\ DP - Z_2T_1}{X - Z_2}}{T_3 - \dfrac{X\ DP - Z_2T_1}{X - Z_2}}$$

$$\frac{XT_1 - Z_1T_1 - X\ DP + Z_1T_1}{XT_2 - Z_1T_2 - X\ DP + Z_1T_1} = \frac{XT_1 - Z_2T_1 - X\ DP + Z_2T_1}{XT_3 - Z_2T_3 - X\ DP + Z_2T_1}$$

Collecting terms and dividing both sides by $(XT_1 - X\ DP)$,

$$XT_2 - Z_1T_2 + Z_1T_1 - X\ DP = XT_3 - Z_2T_3 + Z_2T_1 - X\ DP$$

or, $$XT_2 - Z_1T_2 + Z_1T_1 = XT_3 - Z_2T_3 + Z_2T_1 \qquad (8)$$

But, $0.24\ G(T_1 - T_2) = Q_{S1}$

Therefore, $T_2 = T_1 - Q_{S1}/0.24\ G$; and $T_3 = T_1 - Q_{S2}/0.24\ G \qquad (9)$

Substituting (9) in (8),

$$XT_1 - XQ_{S1}/0.24\ G - Z_1T_1 + Z_1Q_{S1}/0.24\ G + Z_1T_1 =$$
$$XT_1 - XQ_{S2}/0.24\ G - Z_2T_1 + Z_2Q_{S2}/0.24\ G + Z_2T_1$$

[17] Loc. Cit. See Notes 6 and 7.
[18] Loc. Cit. See Note 4.

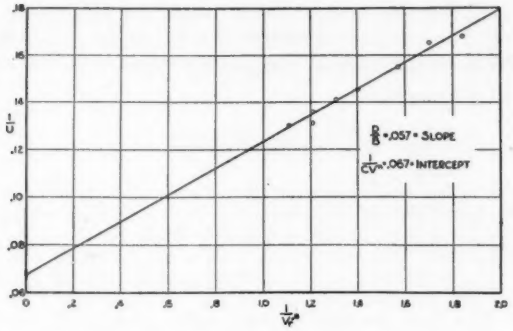

FIG. 5. GRAPHICAL METHOD FOR FINDING FILM COEFFICIENTS (FIRST STEP)

Collecting terms and multiplying by 0.24 G,

$$Z_1Q_{S1} - XQ_{S1} = Z_2Q_{S2} - XQ_{S2} \quad (10)$$

But, $Z_1 = Q_{L1}/Q_{S1}$; $Z_2 = Q_{L2}/Q_{S2}$ and $X = Kdm/dT$ and
$K = L/0.24 \times 7000$

Substituting these values in (10)
$Q_{L1} - (L/0.24 \times 7000)\ (dm/dT)\ Q_{S1} = Q_{L2} - (L/0.24 \times 7000)\ (dm/dT)\ Q_{S2}$

Therefore,

$(L/0.24 \times 7000)\ (dm/dT) = (Q_{L1} - Q_{L2}) / (Q_{S1} - Q_{S2}) =$
$L\ (m_1 - m_2 - m_1 + m_3) / 0.24 \times 7000\ (T_1 - T_2 - T_1 + T_3)$
or, $dm/dT = (m_2 - m_3) / (T_2 - T_3)$ (11)

Now, points 2 and 3 correspond to points D and F in Fig. 4. Therefore, line DF has a slope equal to dm/dT which has been assumed constant and equal to the slope of AB. This assumption is a close approximation for that part of the saturation curve used in comfort air conditioning.

Therefore, it has just been proved that the line connecting point D to point F in Fig. 4 has the same slope as the line AB, connecting the initial and final dew-points on the saturation curve. But D and F have the same dew-points as the initial and final conditions, (A and B). Therefore, a line with the same slope as AB drawn through D must intersect a curve through D and at a constant horizontal distance from the saturation curve, precisely at F. But in the operating range of comfort air conditioning, a line of constant relative humidity almost exactly meets this description. Hence, the curve 2, 3, 4 in Fig. 1 (or the curve DF in Fig. 4) may be taken as a line of constant relative humidity. Since the load ratio line is a line which connects initial and final conditions, it also must pass through F. Therefore, point F can be determined from the intersection of the line of constant relative humidity through D, and it is not necessary to find the slope of the straight line connecting D and F.

Since F was selected as a perfectly arbitrary point, its position can always be found by intersection of a relative humidity line and the load ratio line as just explained. Therefore, a relative humidity line through point D must be the locus of all exit air conditions as the surface temperature is gradually decreased, other conditions remaining constant.

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APPENDIX NO. 2

Derivation of exit dry-bulb equation. (Equation F, Table 1.)

$$Q_S = h_a A(MTD)N$$

$$Q_S = \frac{h_a A(T_1 - T_2)N}{\log_e \frac{T_1 - T_S}{T_2 - T_S}}$$

But, $Q_S = 0.24\ G(T_1 - T_2)$

Therefore, $0.24\ G(T_1 - T_2) = \dfrac{h_a A(T_1 - T_2)N}{\log_e \frac{T_1 - T_S}{T_2 - T_S}}$

Or, $Nh_a A/0.24\ G = \log_e (T_1 - T_S) / (T_2 - T_S)$.

APPENDIX NO. 3

Film Coefficients

The suggested method for finding film coefficients involves the use of two curves shown by Figs. 5 and 6. Both of these curves may be obtained from dry-coil tests only.

Fig. 5 is an application of a method of finding the inside film coefficient as suggested by McAdams. The curve may be obtained from a series of dry tests run at constant air velocity and variable refrigerant velocity. Since $1/U = 1/h_a + 1/h_r$, the following expression may be written by replacing the h values in terms of velocity: $1/U = 1/CV_a^n + R/BV_r^{0.8}$; where C and B are constants to be determined, R is the ratio of external to internal surface, n is an exponent to be determined, and V_a and V_r are the air and refrigerant velocities respectively. Now if tests are run at constant air velocity and variable refrigerant velocity, values of $1/CV_a^n$ will remain constant, and the equation reduces to the general equation of a straight line in which the value of $1/CV_a^n$ will be the zero intercept and the value of R/B will be the slope. (See Fig. 5.)

Now, if tests are run at constant refrigerant velocity and variable air velocity, $R/VB_r^{0.8}$ will be a *known* constant, and the equation will reduce to the following form: $1/U - R/BV_r^{0.8} = 1/CV_a^n$. This is the equation of a straight line on log paper, and the intercept will give the value of $1/C$ while the slope will give the exponent n. (See Fig. 6.)

After Figs. 5 and 6 have been drawn, all the constants in the equation $1/U = 1/CV_a^n$

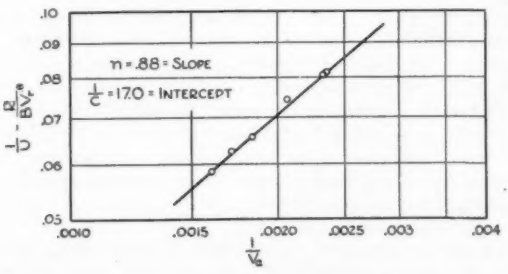

FIG. 6. GRAPHICAL METHOD FOR FINDING FILM COEFFICIENTS (SECOND STEP)

$+ R/BV_r^{0.8}$, will be known, and either the film coefficients or the value of the overall coefficient *for dry cooling* may be calculated for any set of conditions. These dry cooling coefficients may then be applied to cooling in which dehumidification takes place.

DISCUSSION

V. T. KARTORIE (WRITTEN): The humidity method of calculating the performance of extended surface coils is a definite contribution of this paper. However, its greatest value lies not in the suggested procedure, but in the analysis of the basic laws governing the heat transfer from either a dry- or wet-coil. Physical characteristics such as fin design, fin spacing, tube diameter, tube arrangement, coil depth, and the feed arrangement can be accounted for by empirical constants, but the fundamental equations to which these may be applied are still lacking.

Irrespective of whether the form of equation F (Appendix 2) is correct or not, it involves three variables, h_a, N, and T_s (air-side surface coefficient, number of rows of coil depth, and surface temperature, respectively), which are indexes or factors determining the total cooling capacity and operating load ratio of the coil. Because this is so, it would be more desirable to express the cooling capacity in terms of variables which are usually given in the statement of the most frequent problem requiring the selection of a coil size for a given service. These are, entering air conditions, air velocity, refrigerant velocity, refrigerant temperature, load ratio and total load. As an example, for a dry-coil using water circulation, the form is:

$$Q_t = T_e - t_e\left(\frac{C \times \text{gpm}}{12\,C + K \times \text{gpm}}\right)$$

in which the symbols are the same as in Professor Tuve's paper, except:

gpm = total water circulated, gallons per minute.

t_e = entering water temperature.

K = empirical constant (a single value for all coil combinations of same fin design).

C = a function of air velocity, refrigerant velocity, face area, and number of rows of coil depth.

For a direct expansion coil (dry cooling), the cooling water temperature range would disappear and the entering water temperature would be replaced by the temperature of the refrigerant.

When the coil is wet, the heat transfer is an exponential function and the form becomes more complicated. The form for a direct expansion coil is:

$$Q_t = \frac{C}{K}\,(10^{M} - 10^{M'})$$

where: Q_t = total heat transfer.

C = a function of air velocity, refrigerant velocity, face area, and number of rows of coil depth.

K = empirical constant (a single value for all combinations of the same fin design).

$M = 0.0107\ WB_1 + 0.77$ (very nearly).

WB_1 = entering air wet-bulb temperature.

$M' = 0.0107\ T_r + 0.77$ (very nearly).

T_r = refrigerant temperature.

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These equations determine only the cooling capacity, and this must be known before the actual load ratio can be found because the difference between the surface temperature and refrigerant temperature is a function of the total cooling capacity, or vice versa. For a direct expansion coil, the form of the load ratio equation is:

$$\frac{Q_t}{Q_s} = \frac{10^M - 10^{M'}}{0.24\ (T_s - T_r - fQ_t)}$$

in which fQ_t = the difference between the surface and refrigerant temperatures.

When the refrigerant is chilled water, the equation above is modified to include the entering water temperature and the temperature range of the water. Since the most frequent application problem is similar to Example 5 of Table 4, the ratings should be expressed in terms of the variables given there.

The trial and error method is extremely difficult to eliminate, because the constants vary for the numerous arrangements. In the derivation of Equation F (Appendix 2) the value of h_s is multiplied by N to give the sensible cooling capacity. The first equation given in that appendix ($Q_s = h_s \times A \times (\text{mtd}) \times N$) is no doubt correct as is. But the true mtd is not definitely known. Therefore, the surface temperature T_s must be fictitious or imaginary, determined by calculation from the dehumidifying characteristic of the coil under test, unless an accurate means can be found to determine it experimentally. It should be added that while the air-side surface coefficient can be correctly determined because it is based on one degree of mean temperature difference, in application the most difficult to determine is the *actual* mtd over the entire surface.

The results in Table 6 show a few points with very low load ratios, $\frac{(S)}{(T)}$, which are very rare in practice. It is suggested that test be concentrated on load ratios no lower than approximately 0.50 or 0.45, because values lower than that apply to very special applications and, when those conditions arise, frequently reheat or by-pass is required. Also, in the test runs with high load ratios, the air velocity is very low resulting in an extremely high diffusion temperature, which, of course, is to be avoided. Air velocities ranging from 400 to 600 fpm are suggested.

A plus or minus 5 per cent penalty on capacity (due to inaccurate methods of test) is a considerable amount in applications of 50 to 100 tons, and has a very decided effect on the condensing system selection, the resulting brake horsepower per ton, and the compressor motor size. To obtain an accurate balance of the system for the given load, accuracy of equipment selection is extremely important. The calculation of the load itself involves inherent errors which may easily be plus or minus 5 per cent and if the guarantee conditions are to be met, the source of error must be minimized. Therefore, the writer is in entire agreement with Professor Tuve's belief that to attain accuracy of the final method of ratings, the refrigerant and air-side coefficients must be refined. How these are affected by the coil design and arrangement should most certainly have a very important influence on the development of an acceptable and uniform method of rating.

There is little agreement between manufacturers on the air resistance pressures of dry- and wet-coils, and the highest air velocity permissible without the use of eliminator plates to prevent carry-over of entrained moisture into the duct system. These could be observed along with the heat transfer tests.

Professor Tuve is to be complimented on the excellent manner of presenting his fourth paper on this subject. The mathematical approach he has utilized is a significant contribution, which should aid in eliminating temporary expedients and establishing a basic system of laws which govern the performance of extended surface coils.

R. H. SWART[19] (WRITTEN): The tabulation on the various methods of figuring coils with the presentation of the relative advantages and disadvantages was a fine piece of work, and I am only sorry that the method of solution proposed in my article[20] was not listed, too.

The proposed humidity method concept is a very interesting and useful contribution. As they realize, it is difficult to determine point No. 2 on Fig. 1, the DB temperature at which dehumidification starts, by experiment or by calculation because of difficulties in determining what is meant by surface temperature. But even approximate figures are helpful, and I have used it in one calculation upon a very deep core of 20 rows, using water, and finding the surface temperatures of each row, the calculated humidity agrees very well with experimental values. Actually, I have found that, with high air velocities and high temperature gradients through fins and tubes, dehumidification starts when the tube temperature is below the entering air dew-point, and is very appreciable at surface temperatures equal to this temperature.

I was also interested in the Appendix No. 2 and No. 3 wherein the derivation of exit dry-bulb temperature and surface temperatures verifies my own work.

The suggested program for fin tube research is well outlined. There is one question, however. There is no mention made of the well derived formula for determining

$$h_t = h_s \left(1 + 0.62 \frac{C_e - C_s}{T_e - T_s} \right)$$

the relation between the sensible and latent air film coefficient, which I have found to work out well in practice, as also does Pownall. Is the inference in No. 1 suggestion that this is not sufficiently established?

There is no question that research is required for these coefficients, but whether this research can be made general enough to be of value is a question. It seems to me that on the air-side, each design will require a test, as it will be impossible to predict the effect of turbulence, fin corrugations or coating on the factor. On the refrigerant-side, there are so many variables it seems almost necessary to make actual tests, rather than depend upon constants. Even in dealing with coefficients for boiling water, McAdams mentions many surface conditions which affect this, all of which apply to evaporating refrigerants, plus some others such as oil concentration, liquid sub-cooling and per cent of liquid in the cooling mixture. The last sentence in No. 3 is particularly apt, "Suitable experimental methods for dichlorodifluoromethane-coil tests should be developed and specified."

The fourth and fifth suggestions are perhaps something which pertain to the designer and manufacturer more than to general engineering literature. The air friction will always be a test value, and will be part of the specifications of particular designs. Likewise, the effect of depth of cores upon performance can readily be tabulated after the fundamental performance of the coil is known.

This paper is a very interesting contribution to the art, both as a summary of various methods of attacking fin coil problems and as an original method of analyzing exit air conditions from a coil.

H. B. POWNALL[21] AND S. P. SOLING[22] (WRITTEN): We have plotted experimental data which we have available for a coil of constant depth and with constant air velocity for varying refrigerant temperature and have found that when the refrigerant temperature is lowered there is a tendency for the leaving dry-bulb temperature to

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[19] Chief Engr., General Refrigeration Sales Co., Beloit, Wis.
[20] Loc. Cit. See Note 5.
[21] York Ice Machinery Corp., York, Pa.
[22] Engr., Research Dept., York Ice Machinery Corp., York, Pa.

approach the saturation line. It is found that this deviation was slight varying from about 88 per cent relative humidity with a 50 deg refrigerant temperature to approximately a 90 per cent relative humidity with a 30 deg refrigerant temperature. However, the authors state that, "The locus of the exit air conditions from a coil, as refrigerant temperature is varied, is a line on the psychrometric chart at a constant horizontal distance from the saturation curve." Now a constant horizontal distance on the psychrometric chart means a constant number of dry-bulb degrees. In the previous cited example the leaving dry-bulb was 4 deg higher than the dew-point for the 50 deg refrigerant and 3 deg higher than the dew-point for the 30 deg refrigerant.

It is true that these discrepancies are small in this particular range; however, our interpretation of the mathematical proof set forth in Appendix No. 1 shows this variation and also shows that the discrepancy would be larger depending upon the existing condition. For this reason we believe that these limitations of this method

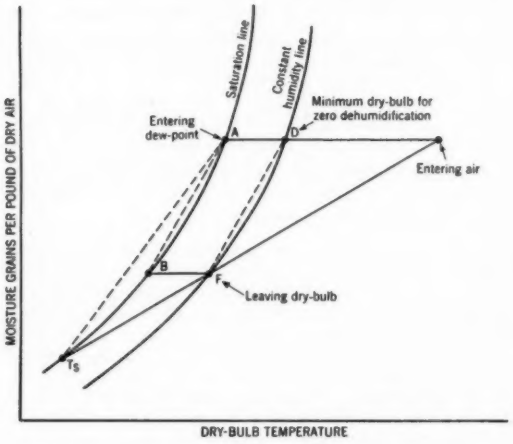

FIG. A. INTERSECTION OF THE LINE *EF* WITH THE SATURATION LINE

should be brought out, and as shown on the authors' Fig. 4 and assumed in Equation 11 AD is not equal to BF.

Referring to Appendix No. 1, in Equation 2, $m_a - m_b = dm/dT\ (T_a - T_b)$, in this equation, the quantity dm/dT is the slope of the line AB since points A and B have been used in establishing this equation. Now in Equation 3, $[Q_1 = \frac{L}{7000}\ (m_a - m_s)\ f(V_a)]$ the authors are no longer dealing with points A and B but with points A and T_s, which is assumed to be the surface temperature. On Fig. 4 of this paper the point T_s is not shown, but would be the intersection of the line EF with the saturation line, which is shown in Fig. A. In Equation 3 the $f(V_a)$ is equivalent to the contact factor as presented in a previous paper.[22]

[22] The Contact Mixture Analogy Applied to Heat Transfer With Mixture of Air and Water Vapor, by W. H. Carrier. (*A.S.M.E. Transactions*, Vol. 58, 1936).

In Equation 4 $[Q_1 = \frac{L}{7000}\frac{dm}{dt}(DP - T_s)f(V_s)]$ it is stated that the quantity $(m_a - m_b)$ has been replaced by substituting its equivalent form from Equation 2. However, this quantity $(m_a - m_b)$ has not been replaced, but the quantity $(m_a - m_s)$, which is the total possible moisture removable assuming cooling to coil surface temperature, has been replaced by its equivalent form from Equation 2. It is necessary to use the difference between the entering dew-point and the coil surface temperature in conjunction with the contact factor $f(V_s)$ in order to arrive at an Equation for T_s which is given as Equation 6.

Therefore, in Equation 4 the slope dm/dt must be the slope of the line At_s, our Fig. A, since the slope in this equation depends upon the actual values of DP and T_s.

In Equation 7 the authors have substituted Equation 5 for T_s, and finally prove that $dm/dt = (m_2 - m_3)/T_2 - T_3$ where $(m_2 - m_3)/T_2 - T_3$ would be the slope of line DF. Now the authors state that dm/dt is the slope of line AB, whereas it is shown that dm/dt is the slope of line AT_s.

This means that the line BF or split between the leaving dry-bulb and dew-point of the leaving air will be slightly less than the difference between the entering dry-bulb and dew-point of entering air, or line AD, since the slope of line DF is the same as line AT_s rather than line AB.

The amount of error that would be introduced by the assumption of equal splits between the air temperature and dew-point for the entering and leaving air would depend upon what part of the saturation curve the problem in question fell. If the straight section of the saturation curve between 40 deg dew-point and 60 deg dew-point is being considered, the error would be small. However, if the curved portion of the saturation curve is under consideration, the error might become relatively large. Therefore, we believe that the statement, "The locus of the exit air condition from a coil as refrigerant temperature is varied is a line on the psychrometric chart at a constant horizontal distance from the saturation curve," should be qualified.

WILLIAM GOODMAN[24] (WRITTEN): Fundamentally, it should be noted that the humidity method described in this paper is based on the assumption that the surface temperature of the coil is constant. This method is, therefore, subject to any errors that are inherent in the constant surface temperature method plus any error due to the assumption that the curve of leaving air conditions coincides with a constant humidity curve. Therefore, the humidity method adds one more approximation to an already approximate method. Whether such benefits as are obtained are sufficient to justify this procedure is an open question.

In Appendix 1 an unjustified assumption seems to have been made in deriving Equation 11. If this is true, then Equation 11 cannot be obtained by the method used in Appendix 1. The derivation in Appendix 1 depends upon the fact that, when the refrigerant temperature is changed, the constant surface temperature of the coil will also change. A different constant surface temperature is obtained for each of the different refrigerant temperatures. However, in deriving the equation in Appendix 1, the assumption has been made that the slope of the line connecting points A and B of Fig. 4 is the same for the different constant surface temperatures that are obtained when the refrigerant temperature is changed. It is apparent that for the same point A, the slope of the line will change as the refrigerant temperature is changed, because a different constant surface temperature must be obtained in each case. Thus in Fig. B which is similar to Fig. 4, one line is shown drawn through points A and T_{s1}, and another line through points A and T_{s2}, where T_{s1} and T_{s2} have the same meanings as in this paper. It is evident from Fig. B that the slope of the two lines is different. For this reason, the value of X (where $X = \frac{dm}{dt}$) in Equation 2 of Appendix 1 must change whenever the value of T_s changes. There-

[24] The Trane Co., La Crosse, Wis.

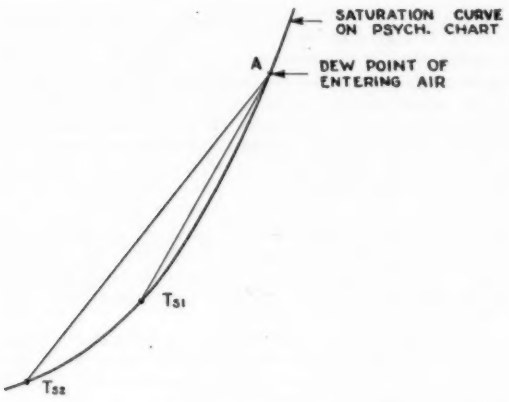

FIG. B. COMPARISON OF TWO SLOPE LINES

fore, if the quantities X_1 and X_2 are substituted in the left and right hand sides respectively of Equation 7, then Equation 11 of Appendix 1 cannot be obtained.

Although the method of obtaining the film coefficients described in Appendix 4 is frequently used, the results obtained by its use are not conclusive. This method depends for its validity upon the assumption that when the points are plotted as in Fig. 5, all of these points will fall closely on a straight line if the exponent 0.8 is used for V_r, whereas if any other exponent is used, the points will scatter more widely. There is no theoretical justification for the exponent 0.8; it is determined by purely experimental and empirical methods. If the data in Fig. 5 are replotted using exponents for V_r which range in value from 0.4 to 1.2, all of the points will also fall on straight lines. In fact, the lower the exponent, the more closely the points will lie along a straight line. Now if different values of the exponent of V_r are assumed, widely varying values of the air and refrigerant film coefficients will be obtained by the method described in Appendix 4. Therefore, as the exponent of V_r must be assumed when using this method, there is no way of knowing which values of the film coefficients are correct. The writer has found in his own work that the use of this method frequently leads to serious error in the determination of the film coefficients.

In Tables 6 and 7 a comparison is given between the relative humidity of the leaving air as obtained by test and the computed relative humidity. A comparison between the total heat removed by the coil as obtained by test and the computed total heat removed would be of interest, as well as a comparison between the sensible heat removed as obtained by test and the computed sensible heat removed. The refrigerant temperatures and the range of initial air conditions used in the tests would also be of interest.

H. F. HUTZEL (WRITTEN): In reading this paper I was very much impressed by the absence of any reference to the temperature gradient between the fins and the refrigerant. All your deductions seem to be based upon the mean effective temperature difference between the air and the surface. I grant that this is correct insofar as the heat transfer is concerned, but this analysis does not enable one to determine the refrigerant temperature required to obtain a given surface temperature. Our experience has been that the difference between the refrigerant temperature and

effective coil temperature may vary anywhere from 4 to 10 deg depending upon the refrigerant, the velocity of refrigerant, and the velocity of air through the coil. Unless one knows what this temperature gradient is, I cannot conceive of applying your formulae to a practical solution of a heat transfer problem.

With reference to method number 7 for determining performance for surface coil dehumidifiers you state in part, on the bottom of page 527, "The air will have a final condition somewhere along the line 2-3-4, which is a curve at a constant horizontal distance from the saturation curve." This seems to be contradictory to a previous claim made, which states that the angle which a straight line connecting the points 1 and 4 on the psychrometric chart makes with the horizontal is a direct measure of the relative amounts of sensible heat and latent heat extracted by the air from the coil. If this is a straight line, there is every reason to believe that the line between points 2 and 4 should be a straight line. We might, for example, conceive of air entering under conditions as indicated by point 2 in Fig. 1. By this reasoning, the line passing between 2 and 4 should be a straight line. If your former claim is correct then we would have no assurance that the line passing through points 2, 3, and 4 would parallel the saturation curve, and unless this is true the point 4, where it intersects the slope line, indicative of the sensible to total heat ratio, would not represent the condition of air leaving the cooling coil.

On account of the many variables involved in the performance of a cooling coil where dehumidification takes place, I cannot believe that a theoretical solution can be offered which would be applicable to all types of coils. In my opinion, a universally applicable equation would have to take into consideration the internal tube diameter, velocity of refrigerant and ratio of extended to internal tube area.

This paper discusses a subject in which I am very much interested. Please do not construe this criticism in the light that this paper is not of value to the industry. On the contrary, it offers some very valuable thoughts which may possibly suggest an accurate solution to this very perplexing problem.

W. L. KNAUS[25] (WRITTEN): By their humidity method the authors show, with good experimental correlation, that the locus of outlet temperature and moisture content follows a line of constant relative humidity for a particular coil with fixed

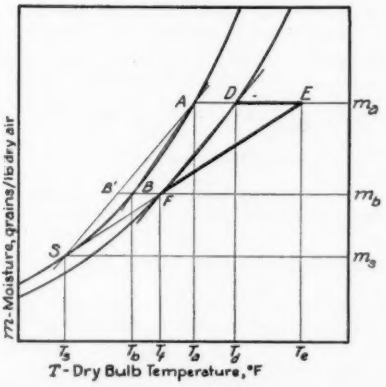

FIG. C. RELATION OF RESULTS WITH SURFACE TEMPERATURE LOCATED

[25] Refrigerating Engrg. Dept., General Electric Co., Fort Wayne, Ind.

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air flow and air inlet conditions, and thus present a simple and useful method of analyzing coil performance. However, there appears to be a slight error in the derivation given in Appendix 1. It probably does not invalidate the conclusion, but may make necessary a review of the analysis and tests to establish the range where the humidity method is satisfactory.

Referring to Fig. C, which is Fig. 4 of the paper redrawn with the addition of the surface temperature point S and the line SA, and to Appendix 2, there may be written,

$$(T_s - T_d)/(T_s - T_a) = (T_s - T_l)/(T_s - T_a).$$

It is seen that the line ES is divided in the same proportion, so that

$$(T_s - T_d)/(T_s - T_a) = EF/ES$$

Therefore triangles EDF and EAS are similar in shape, and line DF is parallel to line AS rather than AB as shown in the paper.

The proof in the paper actually shows this relation. In Equation 2, dm/dT is the slope of line AB. In Equation 4, the following must hold

$$(dm/dT)(DP - T_a) = m_a - m_a$$

For this to be true, dm/dT must be the slope of line AS rather than AB. Hence the derivation following Equation 4 is correct and shows that line DF is parallel to AS rather than AB, as was also shown by the graphical explanation given in this discussion.

The net effect of this correction is to make the displacement BF smaller than AD by the amount $B'B$ and thus cause the locus of outlet conditions to move toward the saturation curve as the surface temperature is lowered. Lines of constant relative humidity show this same tendency to a slight extent and this may be the reason for the good correlation shown with test results.

No. 1100

COOLING REQUIREMENTS FOR SUMMER COMFORT AIR CONDITIONING IN TORONTO

By C. TASKER* (*MEMBER*), TORONTO, ONT., CANADA

A PAPER[1] presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS gave the results of studies made during the summer of 1936 under the Technical Advisory Committee on Comfort Requirements for Summer Cooling at the A.S.H.V.E. Research Laboratory in Pittsburgh, at the Agricultural and Mechanical College of Texas, and at the Ontario Research Foundation, Toronto, Canada.

In brief, the findings were that "comfort may be had over a small effective temperature range, which is independent of daily or even weekly variations in outside weather. The results show clearly that with the weather conditions of July, 1936, in Toronto, an effective temperature of from 2 to 3 deg lower was required for comfort than was the case in Pittsburgh. Similar studies made in Texas prove rather conclusively that for their hotter, more continuous and longer summer heat, the same effective temperature is required as was found for Pittsburgh. In Toronto, 70 per cent of the subjects were found to be comfortable over a range of effective temperatures from 68.5 to 71 deg; in Pittsburgh, from 69.5 to 74 deg; and in Texas, from 71 to 74 deg."

The main criticisms levelled against the results of the studies in Toronto were:

(1) They were based on too small a number of observers, and only male observers were used.
(2) In July, 1936, abnormally warm weather persisted over large parts of North America, and in Toronto an all-time high shade temperature was established.

The results obtained in all the studies were criticized on the grounds that:

(3) Higher age groups might react differently from young, healthy men.
(4) Insufficient data were obtained on the reaction to entering the warm outside weather after being in a cooled space.

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* Research Fellow, Ontario Research Foundation.

[1] Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecki, C. Tasker and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

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The 1937 Technical Advisory Committee on Comfort Air Conditioning (consisting of W. L. Fleisher, vice-chairman, Committee on Research, A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington, R. E. Keyes, A. B. Newton, C. P. Yaglou, and C. Tasker, chairman) agreed that the studies should be continued during 1937 to confirm or modify the con-

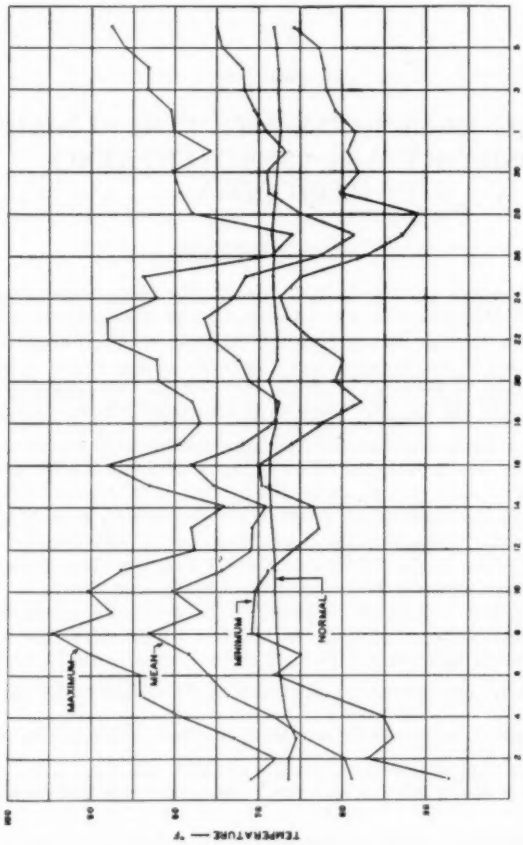

FIG. 1. LOG OF WEATHER CONDITIONS IN TORONTO, JULY 1 TO AUGUST 6, 1937

clusions drawn from the 1936 and earlier studies. The program which was recommended included:

A. Investigations in the offices of a large building located in Minneapolis, Minn., relative to the comfort requirements of male and female employees of various age groups. (One paper [2] describing these tests was presented at the 44th Annual Meet-

[2] Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337).

TABLE 1—SUMMER COMFORT STANDARDS

(Summary of Tests Arranged in Chronological Order)

TEST NO.	DATE (1937)	TIME OF ENTRY E. D. S. T.	RANGE OF OUTDOOR EFFECTIVE TEMP, FOR 2 HOURS PRIOR TO TEST[a] (DEG)	OUTDOOR EFFECTIVE TEMP. ON ENTRY[b] (DEG)	INSIDE EFFECTIVE TEMP. (DEG)	FEELING OF COMFORT AFTER 30 MIN IN TEST ROOM
MEN						
1	July 14	10:30 A.M.	64–67	67.2	71.5	4.5
2	14	2:00 P.M.	68–72	72.2	72.4	5.0
3	15	9:30 A.M.	70–72	72.3	72.9	4.0
4	15	1:45 P.M.	74–77	77.0	72.5	3.8
5	16	10:00 A.M.	72–76	75.4	72.7	4.5
6	16	2:00 P.M.	77–78	77.7	72.1	4.0
WOMEN						
1	July 19	10:20 A.M.	63–68	66.4	71.7	3.8
2	19	2:00 P.M.	70–72	70.7	71.9	4.0
3	20	9:30 A.M.	64–67	68.3	73.7	4.3
4	20	2:00 P.M.	72–74	72.2	73.2	3.8
5	21	9:30 A.M.	64–69	69.0	71.8	4.0
6	21	2:00 P.M.	75–74	73.4	73.0	4.0
7	22	9:30 A.M.	66–72	71.9	74.2	4.6
8	22	2:00 P.M.	77–78	77.6	73.3	4.5
9	23	9:30 A.M.	69–72	72.8	69.4	4.0
10	23	2:00 P.M.	77–78	77.1	70.7	3.7
MEN						
7	July 26	9:30 A.M.	60–62	63.8	70.4	4.0
8	26	2:00 P.M.	64–61	62.0	72.4	4.5
9	27	9:30 A.M.	54–56	56.6	68.3	4.0
10	27	2:30 P.M.	59–61	61.2	71.5	4.3
11	28	9:30 A.M.	54–63	63.0	70.0	4.0
12	28	1:15 P.M.	66–69	67.8	72.1	4.1
13	29	9:30 A.M.	61–67	66.8	71.7	4.0
14	29	2:00 P.M.	71–73	72.5	72.7	4.4
15	30	9:30 A.M.	61–66	66.3	72.1	4.0
16	30	2:00 P.M.	72–73	70.5	71.3	4.0
WOMEN						
11	Aug. 3	10:30 A.M.	66–73	72.1	71.7	4.0
12	3	2:00 P.M.	73–76	74.4	72.4	4.1
13	4	9:30 A.M.	66–70	70.3	72.7	4.0
14	4	2:00 P.M.	75–77	75.2	73.6	4.0
15	5	9:30 A.M.	67–71	71.8	73.7	4.0
16	5	2:30 P.M.	76–79	76.9	70.8	4.0
17	6	9.30 A.M.	67–72	73.7	74.0	4.0
18	6	1:45 P.M.	78–79	77.2	74.1	4.4

[a] Based on measurements made at the Dominion Meteorological Station situated approximately ¼ mile from the building where the tests were conducted.

[b] As measured outdoors in the shade, at the entrance to the building, just prior to the subjects entering. No allowance made for wind.

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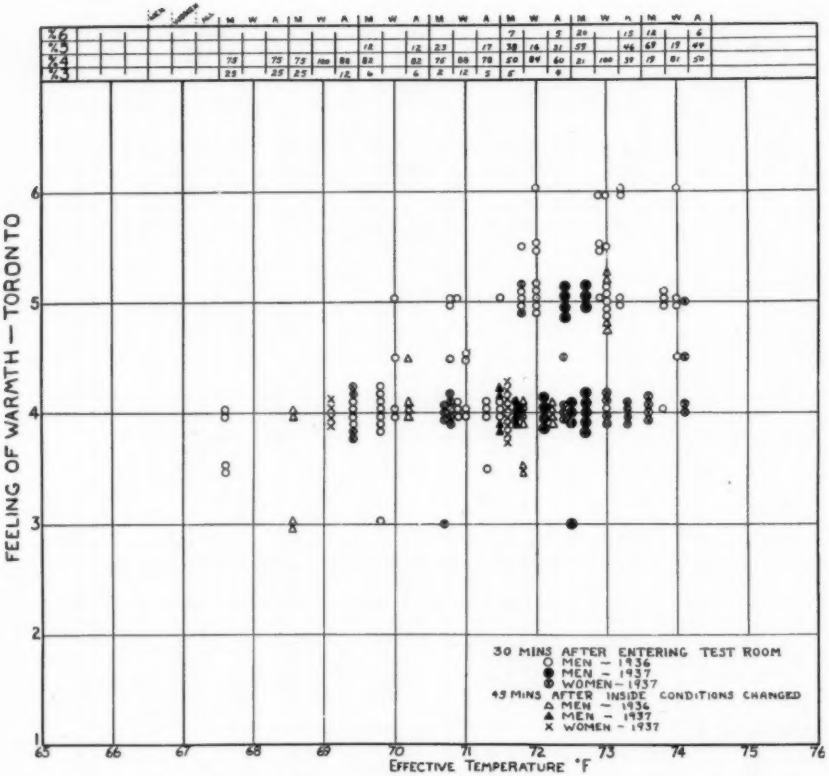

FIG. 2. RELATION BETWEEN EFFECTIVE TEMPERATURE AND FEELING OF WARMTH ON DAYS WHEN OUTDOOR CONDITION WAS WARMER THAN TEST ROOM CONDITION. PERCENTAGE OF EACH NUMERICAL INDEX OF COMFORT GIVEN AT TOP

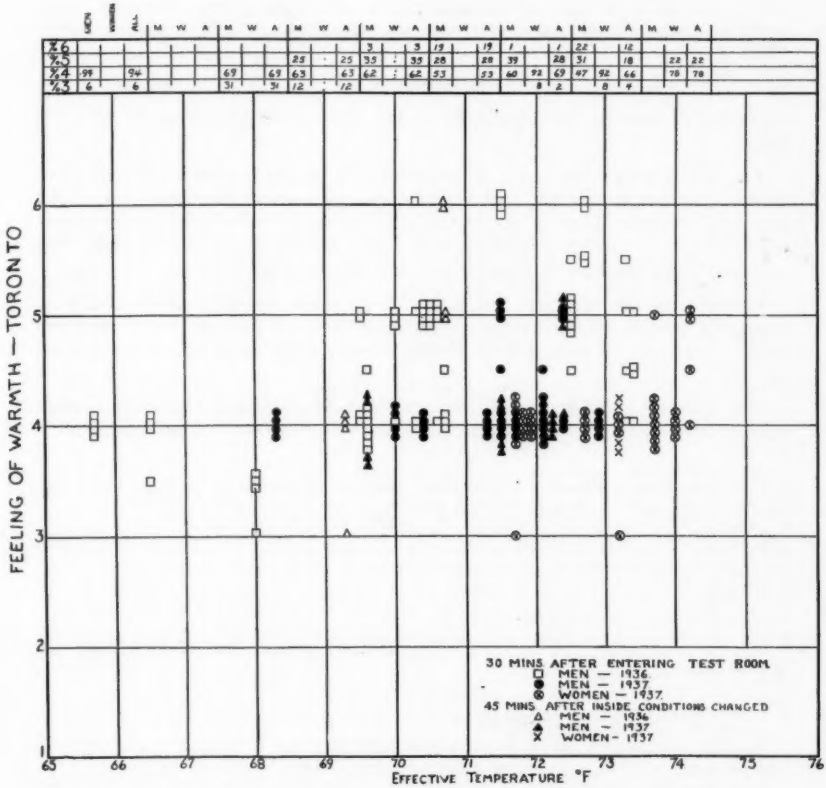

FIG. 3. RELATION BETWEEN EFFECTIVE TEMPERATURE AND FEELING OF WARMTH ON DAYS WHEN OUTDOOR CONDITION WAS COOLER THAN TEST ROOM CONDITION. PERCENTAGE OF EACH NUMERICAL INDEX OF COMFORT GIVEN AT TOP

ing the A.S.H.V.E., and two others are scheduled for presentation at the 1938 Semi-Annual Meeting.)

B. Tests at Texas with particular reference to the transition periods from winter comfort requirements to summer comfort requirements, and vice versa. (The results of these tests are scheduled for presentation at the 1938 Semi-Annual Meeting.)

C. Tests at the Ontario Research Foundation, Toronto, with particular reference to the difference, if any, in the requirements of healthy young men and young women. (This present paper describes the tests made and discusses the results obtained.)

CONDITIONS OF TEST

As far as possible, the conditions of test were similar to those in the 1936 series.

The subjects reported to the Foundation half an hour before each test commenced, and were sent out into the adjacent park where they stayed until the test was started. During the last 8 or 10 min they walked approximately one-third of a mile at a normal pace. On arrival at the Foundation they recorded their feelings of warmth and degree of perspiration.

Immediately on entering the room, they again recorded their reactions, repeating this at 3-min intervals during the first 15 min, at 5-min intervals during the next 15 min, and at 15-min intervals thereafter until the end of the test.

The *scale of comfort* used in these tests is given and it is similar to that used in the 1936 and earlier tests.

4—Comfortable.
3—Comfortably cool—*i.e.*, the subject was not particularly uncomfortable, but would for the sake of comfort desire a warmer rather than a cooler condition.
2—Cool.
1—Cold.
5—Comfortably warm.
6—Too warm.
7—Hot.

At the same time, the degree of sensible perspiration was observed according to the following scale:

0—Forehead or body, dry.
1—Forehead or body, clammy.
2—Forehead or body, damp (perspiration just visible).
3—Forehead or body, wet (sweat covering the surface, frequently in drops).
4—Perspiration on the forehead runs down, or perspiration on the body runs down or wets through clothing.

Since it was desired to investigate *comfortable conditions,* in none of the tests did the degree of sensible perspiration exceed *2*.

The male subjects occupied themselves by reading, clerical work, and general discussions; the female subjects were more industrious, and sewed or knitted during most of the time, while two of them were engaged intermittently in typing. The men tended to become restless if the test exceeded 1½ hours; the women appeared to be at ease even after 2 hours.

There was no control of the type or amount of food eaten by the subjects either at breakfast or at lunch.

SUBJECTS

Pertinent data on the subjects are given in the accompanying tabulations:

MALES

	AGE	WEIGHT	HEIGHT
A	18 years	141 lb	5 ft 11 in.
B	20 years	170 lb	5 ft 11½ in.
C	18 years	148 lb	5 ft 11 in.
D	22 years	148 lb	5 ft 9 in.

FEMALES

	AGE	WEIGHT	HEIGHT
A	26 years	124 lb	5 ft 6½ in.
B	21 years	113 lb	5 ft ½ in.
C	21 years	110 lb	5 ft 1 in.
D	25 years	119 lb	5 ft 5 in.

Two of the male subjects were university students; the other two were well-known local amateur athletes.

All four female subjects were university students.

CLIMATIC CONDITIONS

Fig. 1 shows the climatic conditions in Toronto for the period July 1 to August 6; that is, for 2 weeks previous to the commencement, and throughout the period, of the tests. The daily maximum, minimum, and mean temperatures, together with the daily normal temperature (for 70 years) are given. From Fig. 1 it will be seen that from July 5 to July 11 the daily mean temperature was from 6 to 15 deg above the normal, indicating unusually warm weather.

The first six tests with men were conducted on warm days (July 14-16). Six of the other 10 tests with men were made on days when the mean temperature was from 3 to 10 deg below the normal—*i. e.*, cool days.

Of the first 10 tests with female subjects, eight were on days when the mean temperature was from 2 to 8 deg above the normal; all the other eight tests were made on warm days—*i. e.*, 4 to 7 deg above the normal.

Though most of the tests were conducted on warm days, there was nothing to equal the unusually hot weather experienced during the 1936 tests, when for 16 days the daily mean temperature was from 16 to 24 deg above the normal.

COOLING TESTS AND HEATING TESTS

Table 1 gives a summary of the tests arranged in chronological order.

As in 1936, it was necessary in many of the tests to maintain the test room at a higher effective temperature than the outdoor condition at the time the

subjects entered the room. The results have therefore been split up into four groups:

(1) Tests with men—inside conditions cooler than outdoor.
(2) Tests with men—inside conditions warmer than outdoor.
(3) Tests with women—inside conditions cooler than outdoor.
(4) Tests with women—inside conditions warmer than outdoor.

The results of groups (1) and (3) are plotted on Fig. 2, which also shows the results obtained in the 1936 study with men. Fig. 3 shows the results obtained in the 1937 tests in groups (2) and (4), together with the results obtained in similar studies in 1936.

SUMMARY OF REACTIONS

Table 2 gives in summarized form the reactions of the male and female subjects in the 1937 tests to various effective temperatures. The results are divided into two sections.

A. Reactions 30 min after entering the test room from outdoors.
B. Reactions approximately 45 min after the conditions in the room were changed.

In Table 3 the results of the 1936 studies have been added to those of the 1937 studies with young men. From the data given, the following conclusions may be drawn:

(a) When it was *cooler* inside than outdoors,
young men, normally clothed, were comfortable at 70 to 72½ deg ET.
young women, normally clothed, were comfortable at 69½ to 73 deg ET.
(b) When it was *warmer* inside than outdoors,
young men, normally clothed, were comfortable at 69 to 72 or 72½ deg ET.
young women, normally clothed, were comfortable at 69 to 74 deg ET.

Summarizing the 352 votes cast in the 1936 and 1937 studies by both young men and young women, for indoor effective temperatures of 68 to 74 deg regardless of whether entered from a warmer or a cooler outdoor condition, the accompanying percentages have been found of *3* (comfortably cool), *4* (ideally comfortable), *5* (comfortably warm).

INDOOR EFFECTIVE TEMP.	PERCENTAGE OF VOTES (WARMTH INDEX)[a]		
Deg	*3*	*4*	*5*
68	29	71	..
69	12	80	8
70	2	70	26
71	2	66	23
72	2	64	31
73	3	52	32
74[a]	..	64	33

[a] 75 per cent of the votes cast at 74 deg ET were cast by women subjects.

TABLE 2—SUMMER COMFORT STANDARDS
(*Summary of* 1937 *Tests*)

EFFECTIVE TEMP. (DEG)	FEELING OF WARMTH							
	Indoor Conditions LOWER than Outdoors				Indoor Conditions HIGHER than Outdoors			
	Men		Women		Men		Women	
	A*	B**	A*	B**	A*	B**	A*	B**
68								
68½					4			
69				4−				
69½			4			4−		
70				3+	4			
70½			3+		4			
71			4					
71½		4, 4	4	4, 4	4, 4, 4+, 4+	4	4−	
72	4	4		4	4, 4	4	4, 4	
72½	5, 4, 4+	4	4		4+	4+	4	4
73	5−	4+	4	4	4		4−	4, 4
73½			4+, 4				4+, 4	4
74			4+				4, 4+	4
74½		4+						
75								
75½								
76								

* A—30 min after entering the test room.
** B—45 min after conditions in the test room were changed.

TABLE 3—SUMMER COMFORT STANDARDS
(1936 *and* 1937 *Tests with Young Men Normally Clothed*)

EFFECTIVE TEMP. (DEG)	FEELING OF WARMTH[a]			
	Indoor Conditions LOWER than Outdoors		Indoor Conditions HIGHER than Outdoors	
	1936	1937	1936	1937
66			4	
66½			4−	
67				
67½	3+			
68			3+	
68½	3+			4
69			4	
69½	4−		4, 4, 4	4−
70	4, 4, 4, 4		4+, 5	4
70½			5, 5	4
71	4, 4, 4, 4		4, 4+, 5+	
71½	4	4, 4	6	4, 4, 4, 4+, 4+
72	4, 4, 5, 5, 5	4, 4		4, 4, 4
72½		5, 4, 4+, 4	4+	4+, 4+
73	5, 5, 5, 5, 5	5−, 4+	5, 5+	4
73½			5	
74	5, 5			
74½		4+		

[a] Either 30 min after entering the test room, or 45 min after conditions in the test room were changed.

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COMMENTS AND CONCLUSIONS

From the 1936 tests deductions indicated that young men would be comfortable at 69½-71½ deg ET if they entered from an outdoor condition which was above this range. The 1937 tests seem to indicate that the upper limit of the ideally comfortable zone might be increased slightly to 72 or even 72½ deg ET, but that at 73 deg ET young men, normally clothed, generally felt warmer than desired for comfort.

Young women, on the other hand, were also ideally comfortable at about 70 deg ET, but were comfortable through a wider range, and at 73 deg ET, or even 73½ deg, were not unduly warm.

When the indoor conditions were warmer than outdoor (*i.e.*, at a higher effective temperature), the men in the 1936 tests were comfortable at 69-71 deg ET. The 1937 tests seem to indicate that the upper limit of this range could be extended to 72 deg ET.

For women the limits were not so well defined. Due to lack of time, no tests were made below 71½ deg ET, or above 74 deg. Within this range they seemed to be ideally comfortable with, perhaps, a tendency to desire something a little cooler at 74 deg ET.

ACKNOWLEDGMENT

The author wishes to acknowledge the assistance given by J. Everitt and J. M. McDonell in making these tests and, as chairman of the A.S.H.V.E. 1937 Technical Advisory Committee on Comfort Air Conditioning, to express the Society's thanks to Dr. H. B. Speakman, Director of the Ontario Research Foundation, for making possible both the 1936 series and the present series of tests at no cost to the Society.

(See Joint Discussion, p. 566)

No. 1101

SEASONAL VARIATIONS IN EFFECTIVE TEMPERATURE REQUIREMENTS

By F. E. GIESECKE * AND W. H. BADGETT ** *(MEMBERS)*, COLLEGE STATION, TEXAS

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Texas Agricultural and Mechanical College, conducted under the supervision of the Research Technical Advisory Committee on Sensations of Comfort.

COMFORT requirements for air conditioning have been the subject of extended research by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory and at cooperating institutions since 1923. Early in this work it became apparent that standards of comfort, insofar as temperature, humidity, and air movement are concerned, are not absolute, but are considerably affected by climatic and other conditions. This resulted in the subsequent development of a summer and a winter comfort zone.

These comfort zones at first were assumed to apply universally to indoor air conditioning, regardless of daily variations in the outdoor temperature. In applying these findings to air conditioning in theaters and elsewhere, it soon became apparent, however, that the information available was inadequate and that some consideration should be given to outside temperature. Consequently, there was developed a variable indoor summer comfort standard based upon the outdoor dry-bulb temperature, which was used in the editions of 1933 to 1937 of THE A.S.H.V.E. GUIDE. This standard was based upon a single definite moisture content or dew-point temperature of the air, regardless of the outdoor conditions, and a definite indoor dry-bulb temperature for any given outdoor dry-bulb temperature. Attempts in employing this standard in practice for different air conditioning applications and with different types of equipment resulted in considerable difficulty and dissatisfaction, and the consequent resubmission of the study to the Society's Research Laboratory. In view of the Laboratory's findings in 1935 and 1936, these standards were revised in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1938 and are based upon effective temperature in the conditioned space in relation to the outdoor temperature rather than upon a constant moisture content of the indoor air.

* Director, Texas Engineering Experiment Station.
** Research Assistant, Texas Engineering Experiment Station.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by W. H. Badgett.

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In the Society's summer comfort study in Pittsburgh [1] in 1935 and 1936 it was indicated that an effective temperature of about 72.5 deg was desirable for maximum summer comfort rather than the 66 deg ET generally accepted for winter heating and air conditioning.

Yaglou and Drinker [2] found at Boston that there was an apparent relation between the optimum temperature for comfort which seemed to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained approximately the same value for the months of July, August, and September 1927, and a decrease in the optimum temperature became apparent only when the prevailing outdoor temperature dropped to 66 F, which is below the customary room temperature in the United States for summer and winter. They found that the average optimum temperature for comfort was about 4.5 deg ET higher in summer than that required in winter, a variation which was ascribed partly to difference in clothing worn in the two seasons.

From these various studies it became apparent that there must be some gradual transition from the winter comfort zone to the summer comfort zone, and vice versa, related in some manner to the outdoor temperature, daily, weekly, monthly, or seasonal average. In order to determine this seasonal variation in effective temperature requirements, if possible, a controlled laboratory study was undertaken by the Texas Engineering Experiment Station at the Agricultural and Mechanical College of Texas in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS which ran continuously from January 1937 through May 1938.

These studies were made in an air conditioned laboratory of the Texas Engineering Experiment Station at College Station, Texas. The room was 9 ft wide, 8 ft high, and 18 ft long, with all walls, floor, and ceiling insulated with 3 in. of cork-board. The walls were plastered, and, including the ceiling, were finished with aluminum paint. The floor was 1-in. pine laid over the cork.

The ceiling, which formed the lower side of a plenum chamber, was of rigid insulation made from exploded wood-fiber perforated with 5/16 in. holes about 2½ in. on centers. Into this plenum chamber the conditioned air was introduced and maintained at a slight static pressure which forced it through the ceiling perforations uniformly over the entire test room. Air was returned to the conditioner through two slotted grilles in the ceiling extending across each end of the long side of the room. These slots were arranged so that air was picked up uniformly across the width of the room.

A spray-type air washer, equipped with both cooling and reheating coils, was used to condition the air introduced into the room. A slow speed fan moving about 325 cfm was employed to circulate the conditioned air, and, by means of this slow-speed fan and the use of the perforated ceiling as a distribution grille, a very uniform and imperceptible air movement was obtained throughout the conditioned space, which was well within the meaning of the term *still air*. Fresh air was introduced at a rate of about 8 to 10 cfm per person.

The test procedure was similar to that employed in the previous comfort

[1] Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker, and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145).
[2] The Summer Comfort Zone: Climate and Clothing, by C. P. Yaglou and Philip Drinker (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 269).

zone work of the Society. With 10 male students between the ages of 18 and 23 years, all in good health, serving as subjects, two tests were made each week, one on Thursday afternoon with a group of five subjects, and the other on Friday afternoon with the other five subjects. By using two groups of five subjects each, it was possible to have about 30 sq ft of floor space per subject, thus keeping them far enough apart so that inter-radiation of body heat was reduced to a minimum. The subjects were seated so that each was at least 12 in. from the nearest wall and at least 30 in. from his nearest neighbor. The lighting arrangement of the room was such that the effect of radiant heat from that source was eliminated.

The usual practice was to bring the subjects to equilibrium in a pre-determined constant condition in the test room, which, in each case, was *too warm* for comfort, after which the conditions were made to vary slowly across the comfort zone during a period of from three and one-half to five hours, at a rate never exceeding 1 deg in 8 to 10 min to a condition of *too cool* for comfort, while the subjects recorded their reactions to the conditions at intervals of 5 min according to the following scale of comfort which has been used in most of the Society's previous work on the comfort zone:

1. Cold.
2. Too cool for comfort.
3. Comfortably cool, *i.e.,* the subject was not particularly uncomfortable, but would for the sake of comfort desire a slightly warmer rather than a slightly cooler condition.
4. Ideally comfortable in regard to the sensation of warmth.
5. Comfortably warm, *i.e.,* the subject was not particularly uncomfortable, but would for the sake of comfort desire a slightly cooler rather than a slightly warmer condition.
6. Too warm for comfort.
7. Hot.

At the same time the degree of sensible perspiration was observed according to the following scale:

0. Forehead or body, dry.
1. Forehead or body, clammy.
2. Forehead or body, damp (perspiration just visible).
3. Forehead or body, wet (sweat covering the surface, frequently in drops).
4. Perspiration on the head runs down, or perspiration on the body runs down or wets the clothing.

All subjects were dressed similarly and wore the same clothing throughout all the tests, consisting of the following items: a medium weight wool suit without waistcoat, cotton shirt with collar and tie, cotton undershirt and cotton drawers, both of the type commonly worn in the south the year round, light weight standard length cotton socks, and low-top shoes. The total weight of the clothing worn by each subject, not including shoes, averaged about 4.4 lb, of which the suit averaged about 3.3 lb.

Three separate studies were made of the data secured from these tests. In each case the effective temperature producing optimum comfort for each group for each test was plotted against, first, the average outdoor dry-bulb temperature for the day of the test; second, the average outdoor dry-bulb temperature for the day of the test and the preceding two days; and, third, the average outdoor dry-bulb temperature for the preceding week including the day of the test. The effective temperature producing optimum comfort was considered

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as being the mean of the upper and the lower limits of Zone *4* or the *Ideally Comfortable* zone. These data are shown in Figs. 1, 2, and 3 respectively. Each point represents the average of the reactions of a group of five subjects

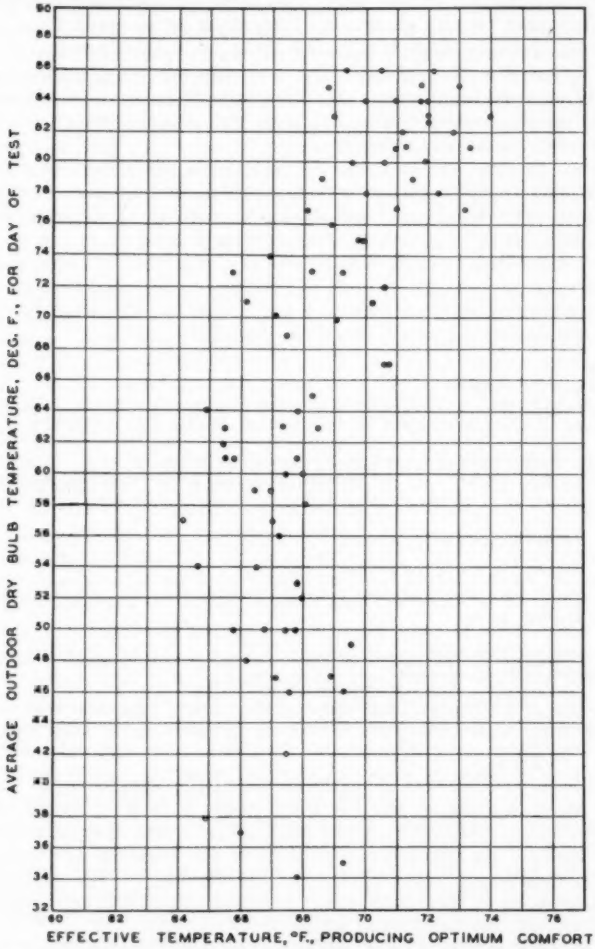

FIG. 1. EFFECTIVE TEMPERATURE PRODUCING OPTIMUM COMFORT PLOTTED AGAINST AVERAGE OUTDOOR DRY-BULB TEMPERATURE FOR DAY OF TEST

for one afternoon's test. If the three figures are superimposed upon each other, they will be found practically identical in their general trend, although the individual points themselves do not coincide. It will be noted that above 65 F a

marked relation is indicated between the outdoor dry-bulb temperature and the effective temperature required for optimum comfort and below 65 F these optimum conditions seem to remain constant or independent of the outdoor

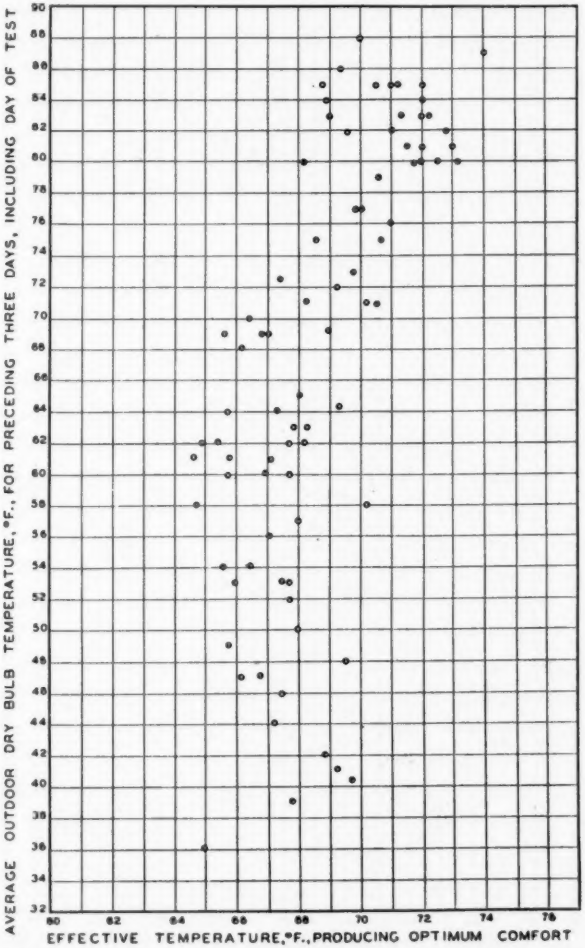

FIG. 2. EFFECTIVE TEMPERATURE PRODUCING OPTIMUM COMFORT PLOTTED AGAINST AVERAGE OUTDOOR DRY-BULB TEMPERATURE FOR DAY OF TEST AND PRECEDING TWO DAYS

dry-bulb temperatures. This change in relationship occurs at the outdoor temperature below which it is generally conceded that heating is required for comfort.

This variation in relationship may be explained as follows: In the winter places of occupancy are closed to the outdoor weather; the greatest part of the time people are indoors and exposed to a more or less uniform temperature;

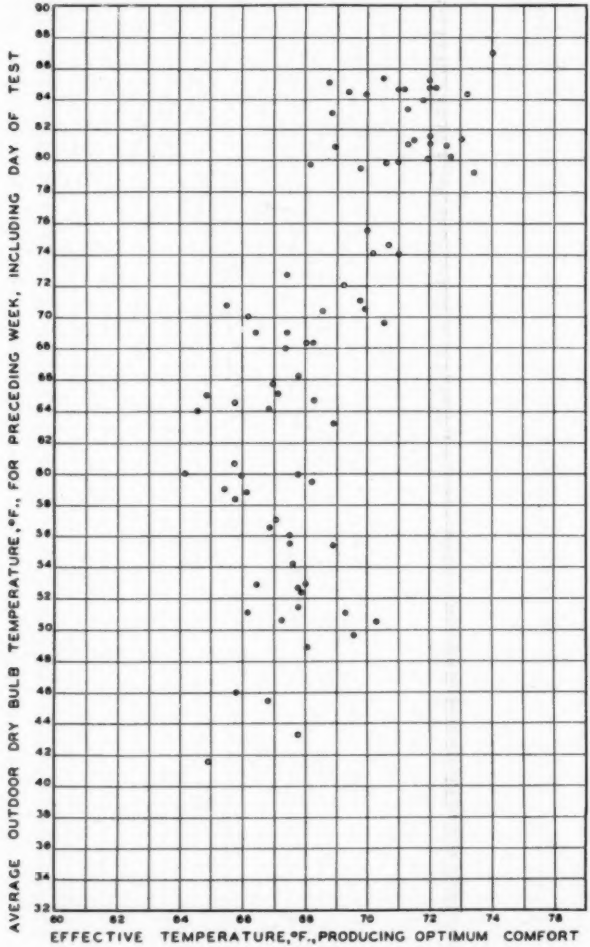

FIG. 3. EFFECTIVE TEMPERATURE PRODUCING OPTIMUM COMFORT PLOTTED AGAINST AVERAGE OUTDOOR DRY-BULB TEMPERATURE FOR THE PRECEDING WEEK, INCLUDING DAY OF TEST

and when they are outdoors, they are dressed according to the prevailing temperature, and, therefore, their response to variations in effective temperature requirements in relation to the outdoor temperature is reduced to a minimum,

and, consequently, winter comfort requirements remain more or less constant throughout the season.

On the other hand, in summer, as it gets warmer, houses are opened and their temperature is allowed to fluctuate with the outdoor temperature; people

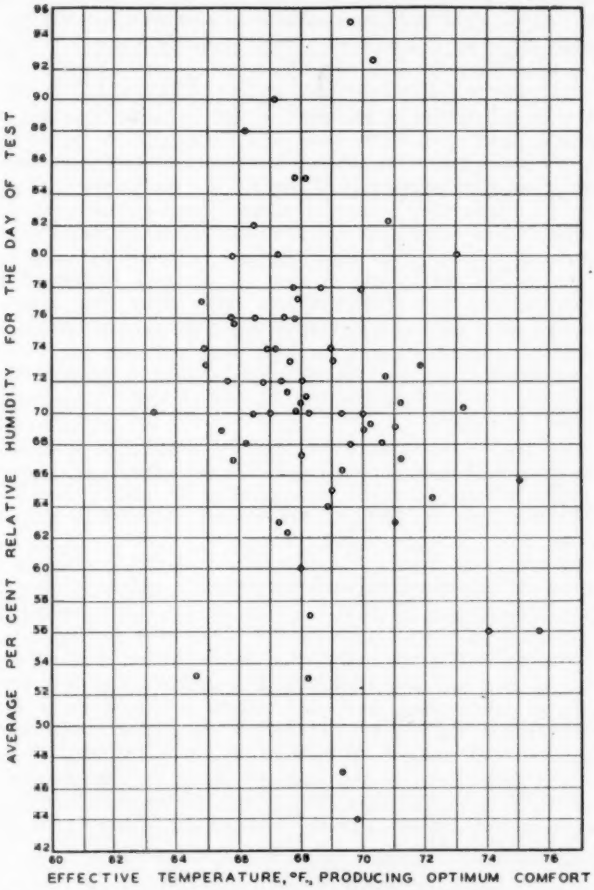

FIG. 4. EFFECTIVE TEMPERATURE REQUIRED FOR OPTIMUM COMFORT PLOTTED AGAINST AVERAGE OUTDOOR RELATIVE HUMIDITY FOR DAY OF TEST

dress as lightly as possible, and, consequently, whether they are indoors or outdoors, their bodies are exposed to temperatures following closely those of the outside, and, therefore, effective temperature requirements vary more closely with the outdoor temperatures.

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In the area of the charts where the average outdoor temperature is above 78 or 80 F, above which temperature summer air conditioning or cooling is generally desired, it will be seen that the upper and lower limits of the comfort zone will extend from about 69 to 74 deg ET, which agrees with the summer comfort zone as previously established for the conditions obtaining at College Station. The optimum condition of maximum comfort for outdoor temperatures above 80 F would vary from about 70.5 deg ET to about 73 deg ET, a mean of about 71.7 deg ET, which is only slightly higher than the previously determined summer comfort line of 71 deg ET for the eastern part of the United States, as published in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, and is the same as the mean of the upper and lower limits of the comfort zone as established for Pittsburgh in the summer of 1936.

The line of optimum comfort of 67 deg ET for the winter comfort zone is only 1 deg ET above the 66 deg ET adopted by the Society as the winter comfort line. These variations between the effective temperature requirements for the North and East and South are so small that they are of no practical importance, particularly in view of the well established fact that comfort requirements vary through a zone rather than along a line.

To determine what relation may exist between the outdoor relative humidity and the optimum comfort requirements, Fig. 4 was drawn. In this chart the effective temperature required for optimum comfort is plotted against the average outdoor relative humidity for the day of test. No correlation between the two is apparent.

A study of the condition at which sensible perspiration disappeared was made and, for the 82 tests, it was found that the average effective temperature at which perspiration ceased was between 75 and 76 deg ET, which agrees with the previous work of the Society establishing this value as about 76 deg ET.

Additional tests were made to determine the variation in effective temperature requirements caused by the coats worn by the subjects during the tests. The finding of the Laboratory, that the wearing of a coat increased the effective temperature requirements approximately 2 deg was verified.

Although these results are based upon determinations secured from only 10 subjects, all men and in good health, it is believed that they are conclusive enough to indicate that there is a definite relation between the indoor effective temperature required for comfort and the outdoor temperature which will be applicable to the general public; that it is indicated that comfort requirements remain more or less constant when the outdoor temperature falls below 65 F, and that above 65 F, the effective temperature required for optimum comfort varies with the outdoor dry-bulb temperature.

JOINT DISCUSSION

(Papers Nos. 1100, 1101, 1102, 1103)

PROF. W. H. BADGETT (WRITTEN): Two additional studies were made, one, Fig. A, in which the effective temperature producing optimum comfort was plotted against the average outdoor effective temperature for the day of test (wind velocity neglected), and the other, Fig B, in which the wet-bulb temperature at the optimum effective temperature was plotted against the average outdoor wet-bulb temperature for the day of test.

These two curves are similar to those of Figs. 1, 2, and 3, with Fig. A, showing the relation between optimum indoor effective temperature and average outdoor effective temperature (neglecting wind velocity), almost identical to these three curves.

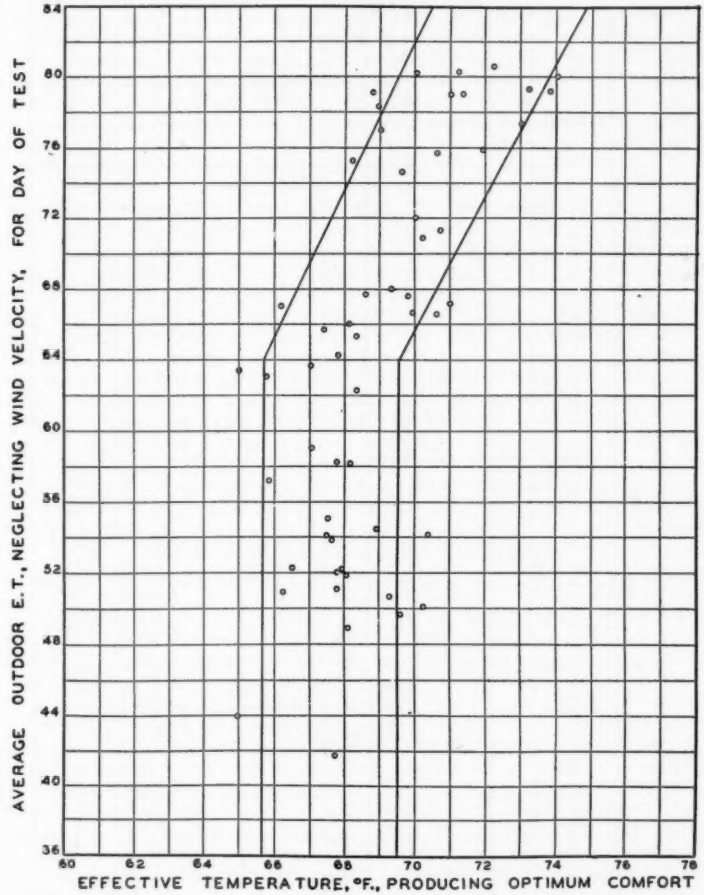

FIG. A. THE RELATION BETWEEN EFFECTIVE TEMPERATURE PRODUCING OPTIMUM COMFORT AND THE AVERAGE OUTDOOR EFFECTIVE TEMPERATURE FOR THE DAY OF TEST (WIND VELOCITY NEGLECTED)

Fig B, the relation of the indoor wet-bulb at the optimum effective temperature to the average outdoor wet-bulb temperature, is similar in trend to the other four curves with the break in the slope occurring about 64 F wet-bulb, but with the limits of the curve several degrees lower on the temperature scale due to the fact that the wet-bulb temperature is lower than either dry-bulb or effective temperature.

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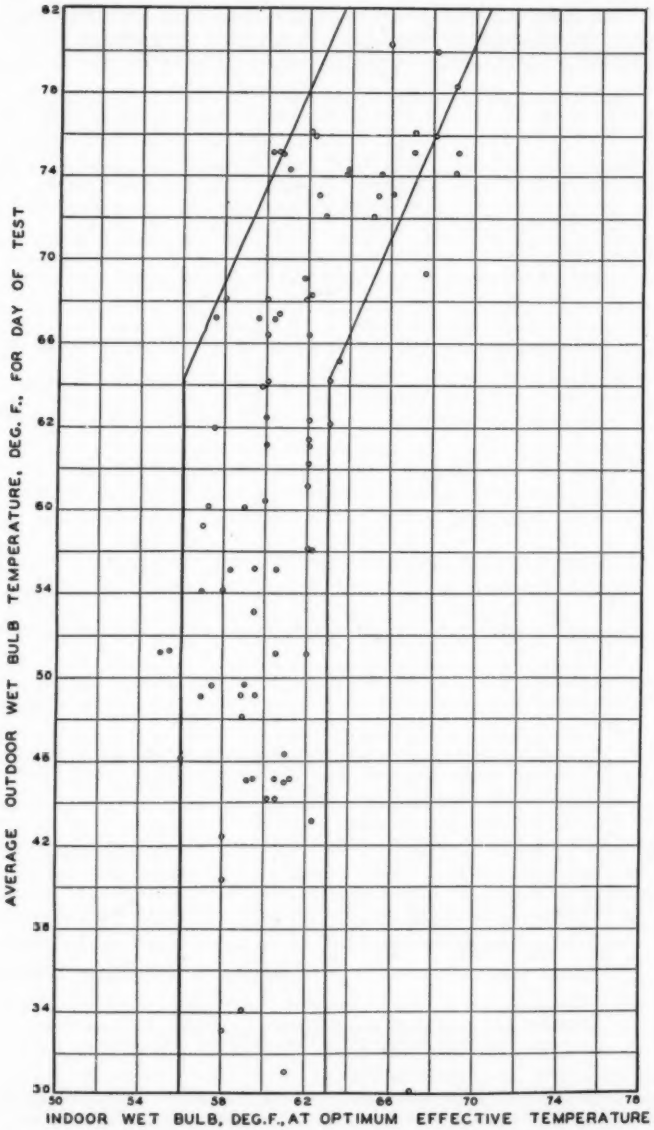

FIG. B. THE RELATION BETWEEN THE WET-BULB AT THE EFFECTIVE TEMPERATURE PRODUCING OPTIMUM COMFORT AND THE AVERAGE OUTDOOR WET-BULB TEMPERATURE FOR THE DAY OF TEST.

L. T. AVERY: The findings in these papers, particularly the paper on Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, have been corroborated in our own work and our observations on the reaction of our own customers to summer conditioning.

We think the criticism of shock has been greatly over-emphasized. No one complains of shock when he leaves an 80-deg office in the winter and walks out on a 10-deg street, and certainly no one complains when he walks on a cold winter's day into a comfortably warm room. Here we are dealing with temperature differences of 50 to 70 deg, or even greater, which compare with the small temperature and humidity differences that exist in summer comfort cooling.

We get no severe reaction when we jump into a swimming pool with a water temperature of 65 to 75 deg, which is probably the maximum in shock due to temperature difference. Shock due to temperature difference is entirely a relative matter. We must keep summer cooling systems within the summer comfort zone so that the person normally occupying the space does not perspire, at least so that perspiration does not accumulate. Criticisms of summer comfort cooling are justified more because of drafts and poor ventilation than because of too low temperatures. We must be particularly careful to maintain summer comfort cooling without sacrificing good ventilation, and without creating excessive air motion called draft.

We need further information on the subject of effective temperature required for different types of work. These papers present the material particularly on office workers who do not sit close together and therefore have no re-radiating influence on each other. It is our opinion that effective temperatures must be carried somewhat lower for theaters or similar locations where people sit close together and where they are somewhat insulated by upholstered furniture.

We have not found the need originally expected for differential control. While it is true that somewhat lower temperatures can be carried when the outside weather is moderate, this seems to result in an excessive waste of cooling effect. An inside effective temperature of 72 to 73 deg is as low as is necessary, and, when the outside weather is cool, this should be accomplished so far as possible by using the outside air. On the other hand we have not found it practical to permit very much rise in inside conditions as 74 deg ET is about the top, above which we get criticism and some people perspire freely. It is our observation that the perspiration line is the important point, and this is affected certainly by the clothing and activity of the persons in the space as well as the effective temperature.

Criticism of summer comfort conditioning frequently comes because the owner of a system wants to make it do tricks. Very often the restaurant owner or theater owner is not satisfied with producing reasonable cooling and dehumidifying, and feels that it is an advantage to advertise and sometimes carry exceedingly low temperatures. I think this is just a phase that we will outgrow by continuously educating our public. Probably when the first winter heating system was installed, the owner ran the temperature up to 90 deg just to prove he could. Gradually more and more people are coming to recognize that good air conditioning means unnoticeable air conditioning, and the proper conditioning system will provide conditions which do not cause comment. We have ceased entirely talking about how warm a house is in the winter time; we take it for granted, unless of course it is overheated. We shall soon take the same attitude toward summer cooling. Most people properly dressed will be comfortable, and these systems should not be run to please the individual who may have a personal preference for colder temperatures or for the owner who thinks it is good advertising to overdo it.

M. S. COOLEY:[a] Regarding the statements of the gentlemen before me, the best air conditioned spaces in Washington are the Chambers of the Senate and House of Representatives. In these rooms the temperature is not much below the temperature outside. However, the humidity is kept at a low point and on entering, the perspira-

[a] Bureau of Yards and Docks, Navy Dept., Washington, D. C.

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tion dries off and there is a feeling of chill for about 2 or 3 min but no shock, thus there is very little discomfort on leaving.

In the drafting room of the Bureau of Yards and Docks at the Navy Department, the temperature and humidity is kept low enough to prevent perspiration on the arms, requiring the protection of drawings by winding tracing cloth around the arms. In this room the temperature is lowered when opening in the morning and the cooling system is shut down about a half hour before closing. In this way those in the room all day leave without shock or discomfort. There are not over a dozen people who enter during the day. In the summer of 1936 there was one day when the outside dry-bulb temperature was 105 deg and the wet-bulb above 70. On this day it was necessary to keep the dry-bulb in the room at 84 deg and the relative humidity below 50 per cent to prevent perspiration.

Among the many visitors that day no one complained of shock. Several of the occupants told me that they were free from hayfever since they had been in the air conditioned room.

Another matter which does not apply directly to the question of comfort is that there seems to be a tendency to overdo in the design of air conditioning. Perhaps we do not know how to figure the heat gain. One system designed by the Bureau was figured by my assistants and checked by me as requiring 11,500 cfm and 19 tons of refrigeration. When advertised, one of the leading air conditioning companies stated that they could not bid as they figured 13,500 cfm and 30 tons of refrigeration were required. They were told that the figures stated were the minimum and that they were required to guarantee the system and to install sufficient capacity to meet the guarantee.

The contractor installed the minimum specified, as this system was for process work. Close control of humidity was required, and the system was operated continuously throughout the year. During the winter there was a complaint of too much air movement in the room, and the air quantity was temporarily reduced to 8500 cfm. Since then we have had outside temperatures of 95 dry-bulb and are still operating with 8500 cfm and 16 tons.

W. E. ZIEBER: These discussions on studies conducted in Toronto, Texas and Minneapolis have so far covered the point of effective temperature and relative humidity and people's shock reactions to them.

The only complaint we have had from serious shock was when the temperature inside was maintained at 70 deg by the purchaser of the equipment, when the outside dry-bulb was 105 deg.

We suggest that these experiments be conducted to include the shock effect on various dry-bulb differences between the inside and outside temperatures.

No. 1102

SHOCK EXPERIENCES OF 275 WORKERS AFTER ENTERING AND LEAVING COOLED AND AIR CONDITIONED OFFICES

By A. B. NEWTON,* F. C. HOUGHTEN ** (*MEMBERS*), CARL GUTBERLET,*** R. W. QUALLEY,† AND M. C. W. TOMLINSON ‡ (*NON-MEMBERS*)

This paper reports the results of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Committee on Research, conducted under the supervision of the Technical Advisory Committee on Sensations of Comfort and analyzed by the Research Laboratory.

RESEARCH into the cooling requirements for summer air conditioning during the past three years has gone a long way toward eliminating differences of opinion regarding desirable temperatures to be maintained in summer cooling and air conditioning for human comfort. Two reports,[1, 2] resulting from studies made on trained laboratory subjects, indicated a need for field studies of a practical nature. A more recent study, made during the summer of 1937 in the Minneapolis offices of a large company, gave a tremendous mass of data on several phases of the subject, including the desirable conditions in offices for persons after one or more hours of occupancy, the so-called *cold shock* upon entering the cooled space from the hot outside, the reaction upon returning to the hot outside after an extended stay in the cooled space, and the general experiences of people working in cooled and air conditioned offices. A paper[3] dealing with the reactions of persons after one or more hours of occupancy in an air conditioned enclosure was recently presented to the Society.

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‡ Statistician, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.

[1] Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215.)

[2] Cooling Requirements for Summer Comfort Air Conditioning by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145.)

[3] Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by F. C. Houghten.

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This paper deals with the cold shock experienced immediately after entering the cooled space from the warm outside, and the hot reactions after leaving the cooled space and returning to the warm outside. It is based entirely upon data collected in the Minneapolis study. A complete descrip-

TABLE 1—DAILY QUESTIONNAIRE TO OFFICE PERSONNEL

Name_____

Location: *Zone 3*

Date: *May 27, 1937*

Insert in Proper Column, "W" for feeling of warmth, "C" for feeling of chilliness.

	IDEAL	MILD	SEVERE
1. Morning reaction to outdoors.	*I*		
2. Reaction upon entering office. Duration: *10 min.*		*C*	
3. Reaction upon leaving for outdoors at noon. Duration: *5 min.* Reaction upon leaving for unconditioned space at noon. Duration:		*W*	
4. Reaction upon re-entering office at noon. Duration:	*I*		
5. General reactions during working hours: Time: 10:30 A.M.	*I*		
3:00 P.M.		*W*	
3:30 P.M.	*I*		
6. Initial reaction during evening. Duration: *Approximately 10 min.*		*W*	
7. Continued reaction during evening.			*W*

8. General remarks: *Sunshine was uncomfortable at 3:00 P. M.*

tion of the office layout, including the air conditioning and cooling systems, and general arrangements leading up to the study appear in the earlier report.[4]

Table 1 is a copy of the questionnaire given to each of the 275 workers in the morning after reporting for work. At the same time the questionnaire filled out for the previous day was returned. This questionnaire indicates that this particular person on May 27, 1937, working in Zone 3 of the building, felt ideally comfortable outside before entering the conditioned

[4] Loc. Cit. See Note 3.

space in the morning, and felt mildly cool for a period of 10 min after entering Zone 3. After leaving the office at noon for the outdoors he felt mildly warm for 5 min. Upon re-entering the office after the lunch period he felt ideally comfortable. At 10:30 A. M., 3:00 P. M. and 3:30 P. M. he felt ideally comfortable, mildly warm and ideally comfortable, respectively. Following his exit from the air conditioned office in the afternoon to the outside, after work, he felt mildly warm for approximately 10 min. Afterwards he had a continued reaction during the evening of severely warm. A dry-bulb temperature and relative humidity recording chart gave the temperature and moisture content of the air in Zone 3 for all periods of that day. Therefore this worker's reactions could be correlated directly with the temperature and humidity of his surrounding atmosphere at any time.

The earlier paper [5] referred to dealt with the data collected under question No. 5. This paper is based on data collected under questions Nos. 2, 3, 4 and 6.

The effective temperature index is logically used in expressing the indoor conditions as affected by the dry-bulb temperature, moisture content and air movement. There is less logic, however, in the expression of the outside conditions by the effective temperature index. The effective temperature, or the index of the feeling of warmth experienced by a person in any environment, should be determined by the dry-bulb temperature, moisture content, air movement and radiation. The effect of the first three of these factors has been evaluated. The effect of radiation has, however, never been evaluated as applying to the effective temperature scale.

Since wind velocity and sun radiation are very important factors in affecting a person's feeling of warmth on the outside, no application of the effective temperature index not including these factors can adequately define the environmental conditions as affecting a person's feeling of warmth. Obviously, the air velocities on the outside affecting any individual worker before entering could not be determined in this study; neither could the sun radiation intensity be evaluated and, as stated previously, its effect on the effective temperature could not be determined even if the sun intensity were known. Hence, only the dry- and wet-bulb temperatures could be used in evaluating the outside effective temperature. While it is admitted that the dry- and wet-bulb temperatures alone constitute a relatively poor index of the feeling of warmth that a person would experience, it is felt that this index is, in some measure at least, better than the dry-bulb temperature alone, and it has therefore been used in this study. In order to indicate the variations in dry-bulb which may be had at a given effective temperature, for the convenience of persons accustomed to thinking in outside dry-bulb temperature, the resulting dry-bulb for the given effective temperatures at three different relative humidities is also given some consideration in this paper.

The study was started on May 20, 1937 and, with the exception of a brief interruption from June 25 to July 6, continued until the end of September, 1937. Two hundred seventy-five workers in all were included in the study, with an average daily attendance of better than 90 per cent. Hence, approximately 22,000 questionnaire cards were collected and analyzed. In order to minimize the complications and opportunity for confusion in filling out the questionnaire,

[5] Loc. Cit. See Note 3.

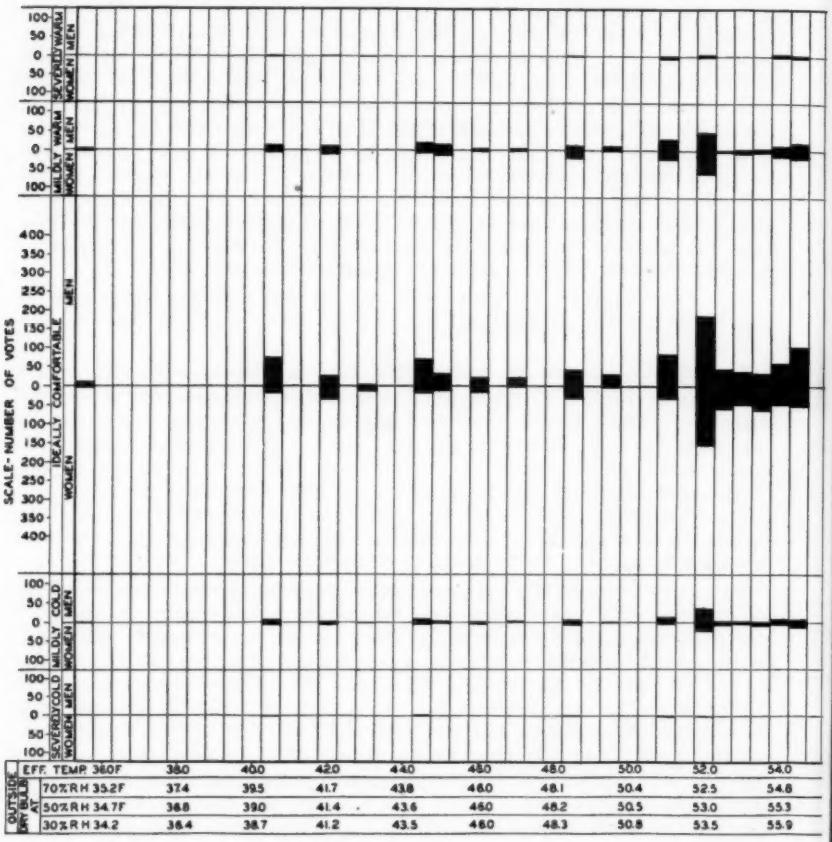
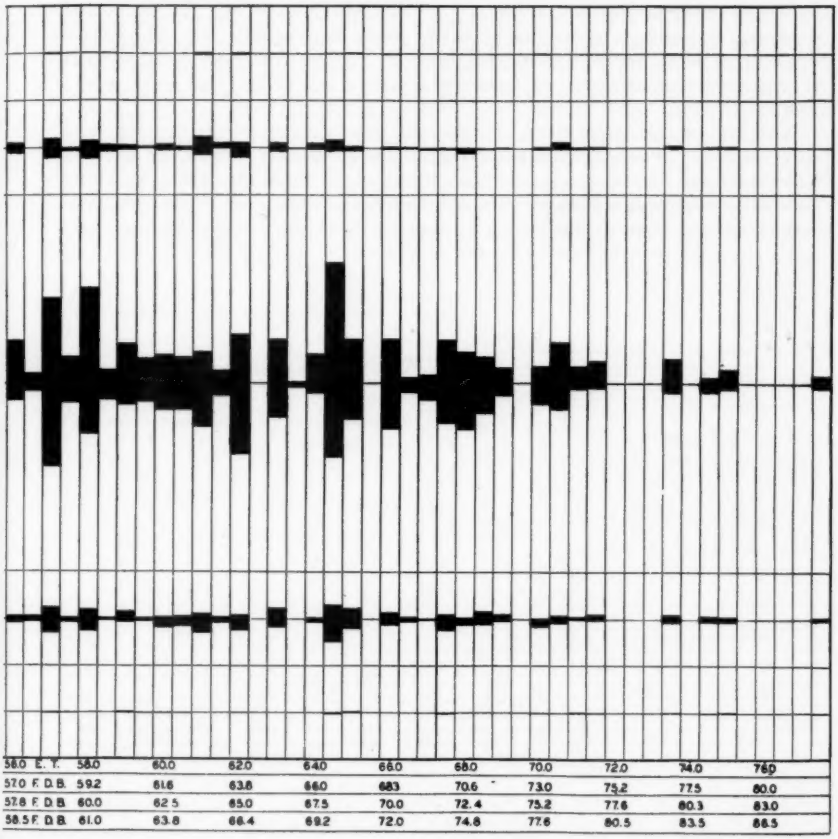


FIG. 1. RELATION BETWEEN OUTSIDE EFFECTIVE TEMPERATURE AND THE ACCOMPANY-MEN AND WOMEN EXPERIENCING VARIOUS FEELINGS OF WARMTH

the scale of feeling of comfort ranging from 1 to 7 used in some earlier Laboratory studies was shortened to include 5 gradations of comfort, namely *severely cold, mildly cold, ideally comfortable, mildly warm* and *severely warm.*

SHOCK FOLLOWING ENTRY INTO CONDITIONED SPACE

A sample of the tabulated data from all questionnaires answered by individual workers upon entering from the out-of-doors at the prevailing weather to the indoor conditioned space, when an indoor effective temperature of from 69 to 70 deg was maintained, is shown in Fig. 1. The plot shows the number of



ING DRY-BULB FOR 30, 50 AND 70 PER CENT RELATIVE HUMIDITY, AND THE NUMBER OF UPON ENTERING THE INSIDE CONDITION AT 69 TO 70 DEG ET

men and women indicating the five gradations in feeling of warmth, in accordance with the accepted scale, immediately following entry to the air conditioned space when this space was kept at from 69 to 70 deg ET. The outside effective temperature from which entry to the conditioned space was made is given by the horizontal axis. Data for each degree range of effective temperature maintained in the conditioned space from 66 to 76 deg, of which Fig. 1 is an example, were plotted in 11 charts.

The mass of data is too great for any accurate deductions regarding the variations in the feeling of warmth experienced for entry from various outside effective temperatures ranging from 36 to 80 F. However, it may be observed

that with lower outside temperatures, such as from 36 to 50 deg ET, there was a greater percentage of feeling of *mildly warm* than *mildly cool,* while upon entering from higher outside temperatures, as from 60 to 80 deg ET, there was a greater percentage who experienced a feeling of *mildly cool* than *mildly warm.*

The data plotted as numbers of individual observations in Fig. 1 are plotted as percentages in Fig. 2. The percentage points for various feelings of comfort upon entering from various outside temperatures show the usual variation found for comfort data on persons. However, the chart does show some consistent relation between the feeling of warmth experienced upon entering from various outside temperatures with the conditioned space kept at from 69 to 70

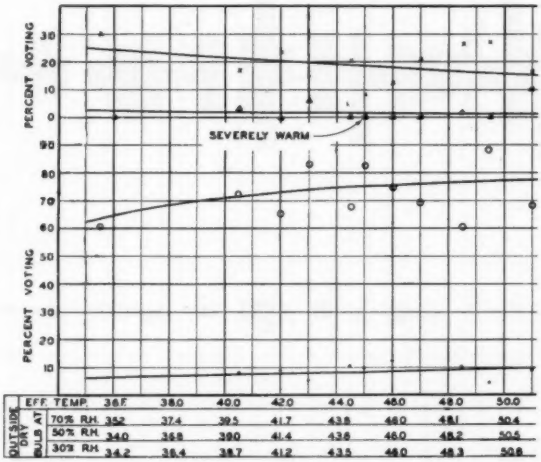


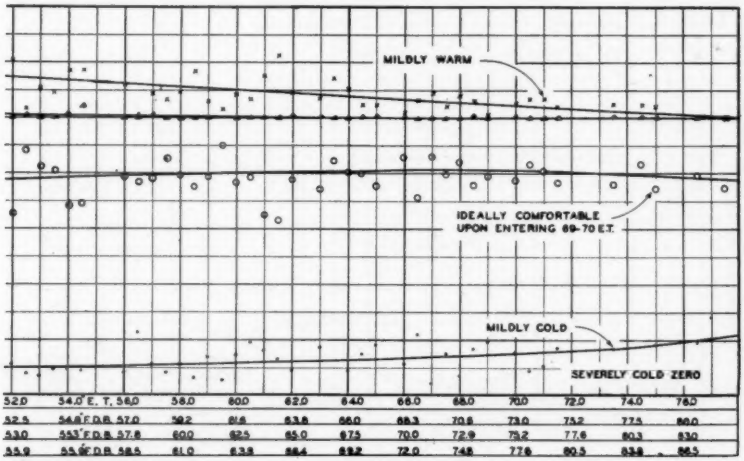
FIG. 2. RELATION BETWEEN OUTDOOR EFFECTIVE TEMPERATURE AND FEELING OF WARMTH UPON ENTERING INSIDE CONDITIONS RANGING FROM 35 TO 78 DEG ET

deg ET. There is a slight falling off in the percentage of comfort for lower temperatures due to an increase in the feeling of *mildly warm* and *severely warm* for lower outside temperatures, and a slight falling off in the percentage of comfort due to an increase in the feeling of coldness upon entering from higher outside temperatures. These variations are not great,—probably for the reason that the sample chosen is for a moderate inside effective temperature of 69 to 70 deg in the entire heating and cooling range from approximately 64 to 75 deg.

From the 11 charts of which Fig. 2 is an example, reactions upon entering the inside maintained at various effective temperatures with relation to given outside temperatures were plotted. Figs. 3, 4 and 5 are examples of a total of 13 such plots. A comparison of the comfort reactions in these three charts for days when the outside temperature was 36, 60 and 77.5 deg ET, respectively, shows marked variations. When entering the conditioned space from the outside at 36 deg ET the occupants were most pleased with their sensations of

warmth if the inside was maintained at 66 deg ET, or lower. As the effective temperature maintained inside rose from 66 to 73 deg the feeling of ideal comfort decreased from 84 per cent to 58 per cent, indicating a higher percentage of people ideally comfortable at effective temperatures below 66 deg. At 66 deg ET 16 per cent of the observations were *mildly cold,* and there were no observations of either *mildly warm* or *severely warm.* With an increase in the inside temperature the percentage of *mildly warm* increased and for higher temperatures a feeling of *severely warm* became significant.

Persons entering various inside effective temperatures in conditioned space from 60 deg ET outside, as plotted in Fig. 4, show considerable contrast with Fig. 3; the highest percentage of comfort upon entering the conditioned space



occurred when the inside effective temperature was from 68 to 70 deg. A higher percentage indicated *mildly cold* and a small percentage *severely cold* for the lower inside temperatures, while a higher percentage felt *mildly warm,* or *severely warm* for the higher inside temperatures given in the chart. Fig. 5, giving the reactions upon entering the conditioned space at various effective temperatures from an outside at 77.5 deg ET, shows considerable contrast from the reactions given in either Fig. 3 or 4. A maximum percentage of comfort was obtained between 70 and 71 deg ET indoors, but there was no indication of *severely warm* or *severely cold.*

In order to give a more ready comparison of the change in the maximum percentage of indications of *ideal comfort,* a series of curves for persons entering inside effective temperatures ranging from 66 to 76 deg, from several outside temperatures ranging from 36 to 80 deg, is given in Fig. 6. There are several interesting facts to be noted in the relationship of these several curves. For low inside effective temperature conditions ranging from 68 to 66 deg,

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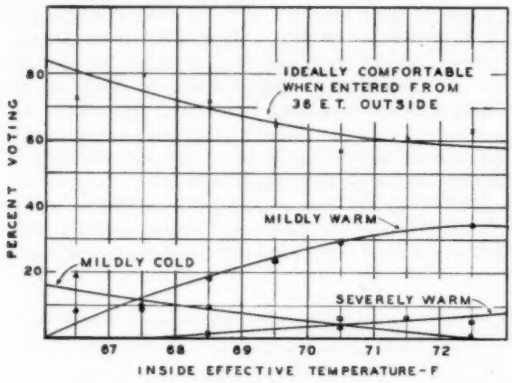


FIG. 3. RELATION BETWEEN FEELING OF WARMTH UPON ENTERING INSIDE CONDITIONS OF VARIOUS EFFECTIVE TEMPERATURES FROM AN OUTSIDE CONDITION OF 36 DEG ET

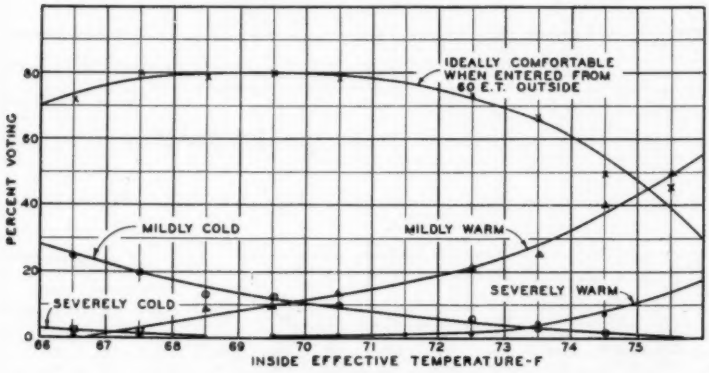


FIG. 4. RELATION BETWEEN FEELING OF WARMTH UPON ENTERING INSIDE CONDITIONS AT VARIOUS EFFECTIVE TEMPERATURES FROM AN OUTSIDE CONDITION OF 60 DEG ET

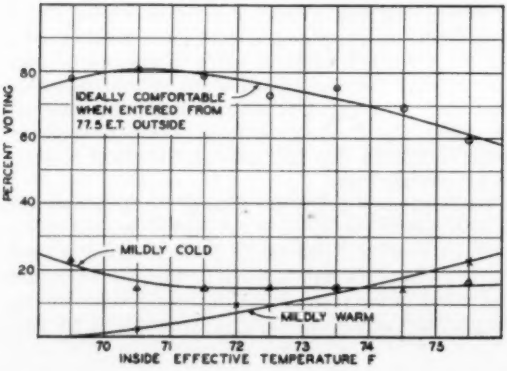


FIG. 5. RELATION BETWEEN FEELING OF WARMTH UPON ENTERING INSIDE CONDITIONS OF VARIOUS EFFECTIVE TEMPERATURES FROM AN OUTSIDE CONDITION OF 77.5 DEG ET

there is a progressive fanning out of curves for outside effective temperatures ranging between 36 to 75 deg. The curve for an outside condition of 80 deg is not continued to lower inside effective temperatures, since these lower inside temperatures were never maintained when the outside effective temperature was 80 deg.

For inside effective temperatures in the neighborhood of 70 deg, there is an almost complete reversal in the order of the curves for outside weather conditions ranging from 36 to 75 deg ET. For higher inside effective temperatures an outside condition of 70 deg ET gives a maximum percentage of comfort with a falling off in this percentage for both higher and lower outside temperatures.

It is of interest to note that curves for outside effective temperatures between 60 and 70 deg give the broadest crest. In other words, less discomfort was

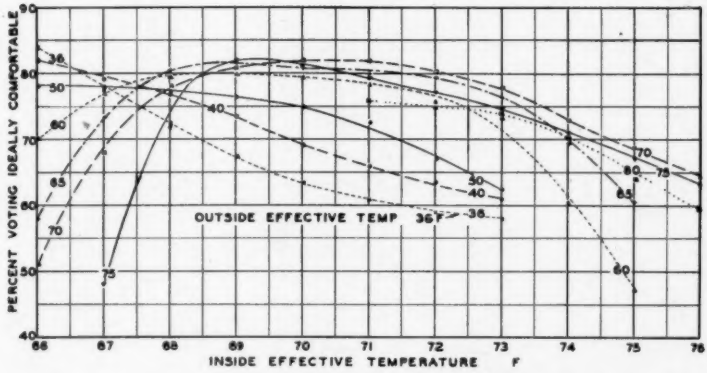


FIG. 6. RELATION BETWEEN FEELING OF WARMTH UPON ENTERING INSIDE CONDITIONS AT VARIOUS EFFECTIVE TEMPERATURES FROM OUTSIDE CONDITIONS GIVEN BY THE SEVERAL CURVES

felt upon entering from outside effective temperatures within the 60 to 70 deg range when a relatively broad range of inside conditions was maintained from approximately 68 to 72 deg. It is also of interest to note the rather complete separation of all the curves into two groups. One group includes entry from outside temperatures of 50 deg and below, where the maximum percentage of *ideally comfortable* is at or below (to the left of) 66 deg ET, and the other group includes entry from outside temperatures of 60 deg or higher. In the latter case the maximum percentage of *ideal comfort* was registered for inside temperatures at or above approximately 69 deg. This would seem to indicate a definite separation of the reactions of persons entering into a heated and air conditioned space from the outside at below 60 deg, from those entering either unheated or cooled and air conditioned space when the outside is below 60 deg, rather than a gradual gradation from the one step to the other.

A composite chart showing the percentage of the various indices of comfort upon entering the conditioned space kept at several different effective temperatures is given in Fig. 7. The curves for *ideal feelings of comfort*

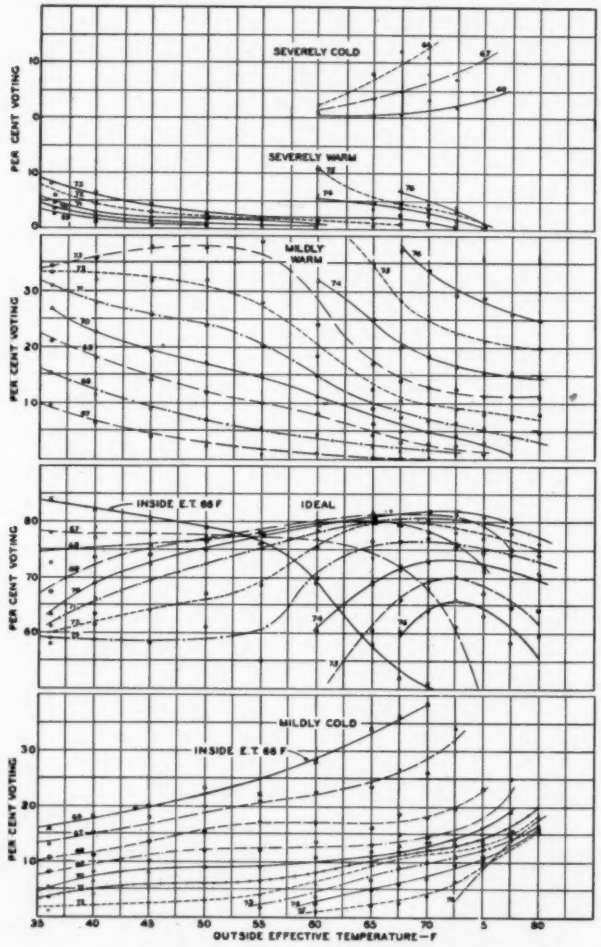


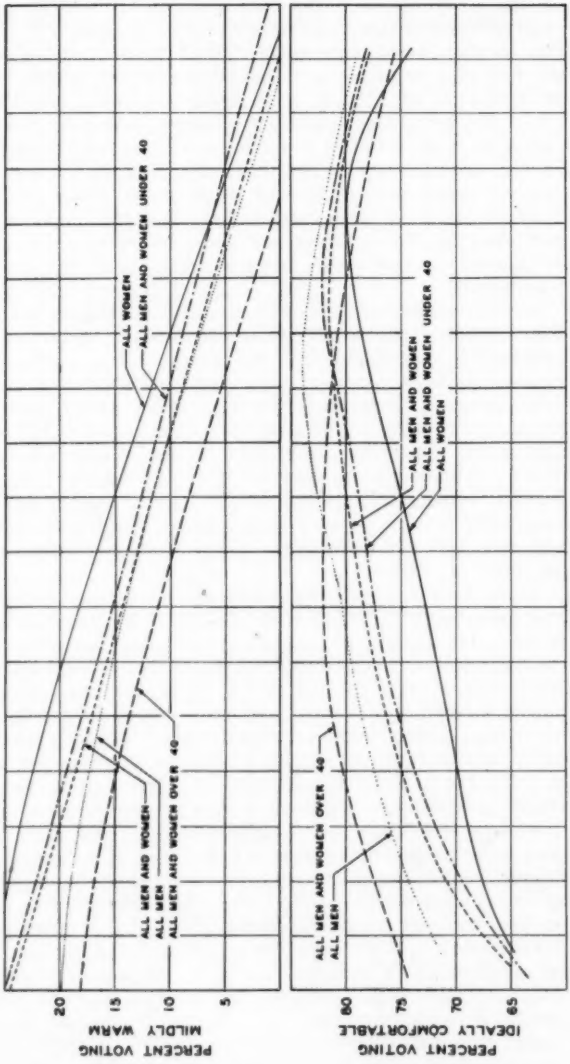
FIG. 7. RELATION BETWEEN OUTSIDE CONDITIONS AND THE FEELING OF WARMTH UPON ENTERING INSIDE CONDITIONS AT EFFECTIVE TEMPERATURES GIVEN BY THE SEVERAL CURVES

for various inside conditions again show a rather critical separation between 50 and 60 deg ET outside. It also indicates a maximum percentage of comfort at 66 deg or lower indoors when the outside temperature was below

53 deg, and maxima of approximately 69, 70 and 71 deg ET indoors for relatively hot outside conditions. In this connection it is of interest to note that in the Minneapolis study it was found that for extended occupancy the largest percentage of occupants were comfortable in the neighborhood of 70 to 71½ deg ET. It is of further interest to note that there is a falling off in the percentage indicating *ideally comfortable* at indoor effective temperatures in the comfort range 69 to 72 deg for outside temperatures above 75 deg, and also to note the progressive drop in the maximum percentage of comfort when outside effective temperatures above 72 deg are maintained. Furthermore it is interesting to note that for higher indoor effective temperatures than 72 deg there is a slight increase, at which the maximum percentage of comfort occurs upon entering the cooled space from the outside temperature.

It will be observed that each successive plotting of the data from Fig. 1 to Fig. 7 smooths out or effectively averages the variations resulting from differences in individuals, with the result that a consistent set of curves is obtained in Figs. 6 and 7. That this averaging gives a true picture of the relationships is indicated by the consistency of the different curves in Figs. 6 and 7. If insufficient data were available for averaging it would be expected that a less consistent relationship in the location of various curves would follow. This consistent relationship should therefore add greatly to the confidence of the results of the study. In this connection, it should be pointed out that in former studies in which only a few trained individuals participated and relatively few observations were made, no such consistent results were or could be obtained.

Three factors govern the choice of an optimum inside air condition for any given outside condition as determined by the cold shock upon entering, namely, the percentage of occupants feeling *ideally comfortable*, the percentages feeling *mildly cold*, and *severely cold*, and the percentages feeling *mildly warm* and *severely warm*. The condition which gives a slightly higher percentage of *ideally comfortable* but at the expense of a relatively high percentage feeling *mildly* or *severely cold* is probably not as desirable as some other indoor condition giving a slightly lower percentage of *ideally comfortable* but a better balance between those feeling too warm or too cold. There is room for argument regarding the relative undesirability of two conditions which give the same percentage of too warm and too cold. Probably a consensus of opinion would favor a condition giving a somewhat higher percentage of too warm accompanied by a somewhat lower percentage of too cold. However, there are insufficient facts available on which to base any such conclusion at this time. As an example, when it is 50 deg ET outside, the *ideal comfort* curves, Fig. 7, indicate that the highest per cent of comfort, or 79 per cent, would be obtained with 66 deg ET maintained indoors. Nevertheless it should be observed that the remaining 22 per cent would all be *mildly cold*. An increase in the indoor condition to 70 deg would reduce those indicating *ideal comfort* to about 75 per cent, but it would at the same time reduce the percentage of *mildly cold* to nine per cent, with an increase in the feeling of *mildly warm* from zero to 17 per cent. From this it may be argued that 70 deg ET indoors would be the most satisfactory condition as regards sensation of comfort immediately upon entering from an outside condition of 50 deg ET. These indications suggest the desirability



of maintaining an indoor condition in the upper range of the comfort zone for any individual in order to eliminate the shock effect.

There is considerable argument in the practical application of air conditioning for definitely increasing the indoor effective temperature with rise

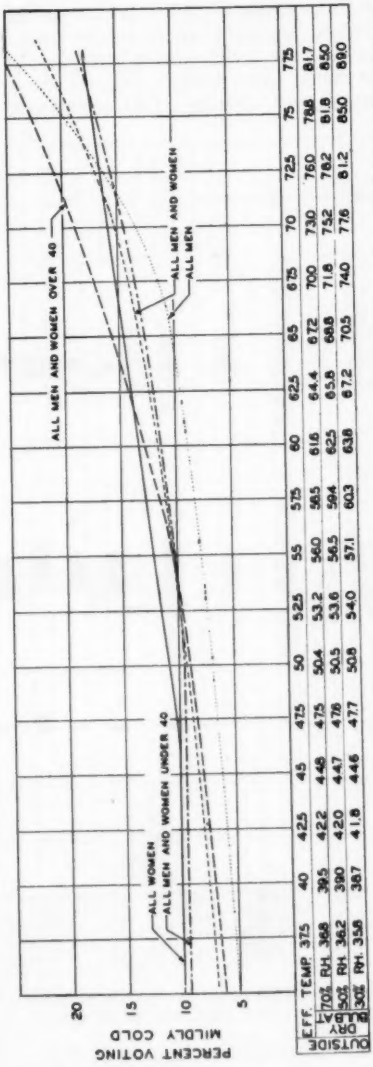


FIG. 8. RELATION BETWEEN PER CENT FEELING OF WARMTH EXPERIENCED AFTER ENTERING THE OFFICES CONDITIONED AT FROM 69 TO 70 DEG ET AND THE OUTSIDE EFFECTIVE TEMPERATURE FOR ALL EMPLOYEES INCLUDING BOTH MEN AND WOMEN OF ALL AGES, ALL MEN, ALL WOMEN, ALL MEN PLUS WOMEN OVER 40 YEARS, AND ALL MEN AND WOMEN UNDER 40 YEARS

in the outside temperature, in order to avoid excessive temperature differentials and thereby eliminate shock effect upon entering cooled space. While a tendency in that direction is indicated by the current study, the deductions from the curves in Fig. 7 must be that very little shock effect is experienced when the inside effective temperature is maintained at the highest level of maximum percentage of comfort. The curves indicate that in the Minneapolis study optimum comfort was not obtained at above 73 or 74 deg ET. For example, the limits of the data collected as far as high outside temperature is concerned indicate no measurable percentage of shock sensation of *severely cold* upon entering an indoor effective temperature higher than 69 deg. There was a relatively small percentage of sensations of *mildly cold* for effective temperatures of 71 or 72 deg, the conditions for optimum comfort, after one hour's occupancy as found in the Minneapolis study. The importance of not maintaining too high an effective temperature (Fig. 7) is also indicated by the rapid drop in percentage of *ideal comfort* when the indoor effective temperature exceeds 72 deg.

RELATIVE SHOCK EXPERIENCED BY MEN AND WOMEN AND BY DIFFERENT AGE GROUPS

In order to determine the relative shock experienced by men and women, and by different age groups of both men and women, some of the data were analyzed for these differences. The percentage of different age and sex groups indicating *ideally comfortable, mildly warm* and *mildly cold,* when

entering an indoor condition ranging from 69 to 70 deg ET from the prevailing outdoor conditions, is plotted in Fig. 8. When entering from an outside condition of 40 to 45 deg, approximately eight per cent fewer women were *ideally comfortable,* approximately four per cent more were *mildly cold,* and about six per cent more were *mildly warm* in comparison with men. After entering

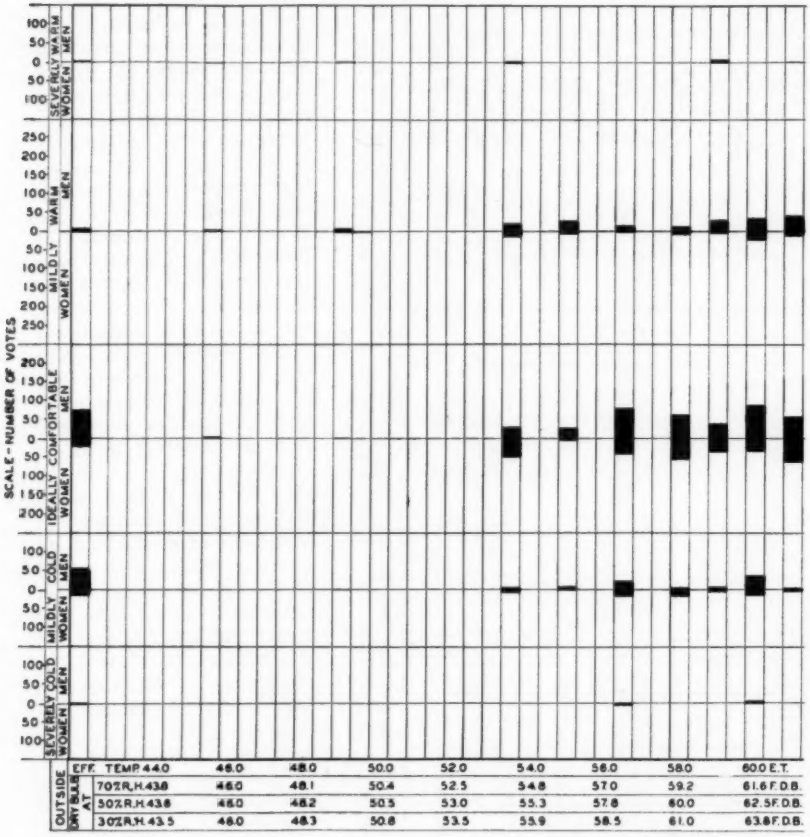
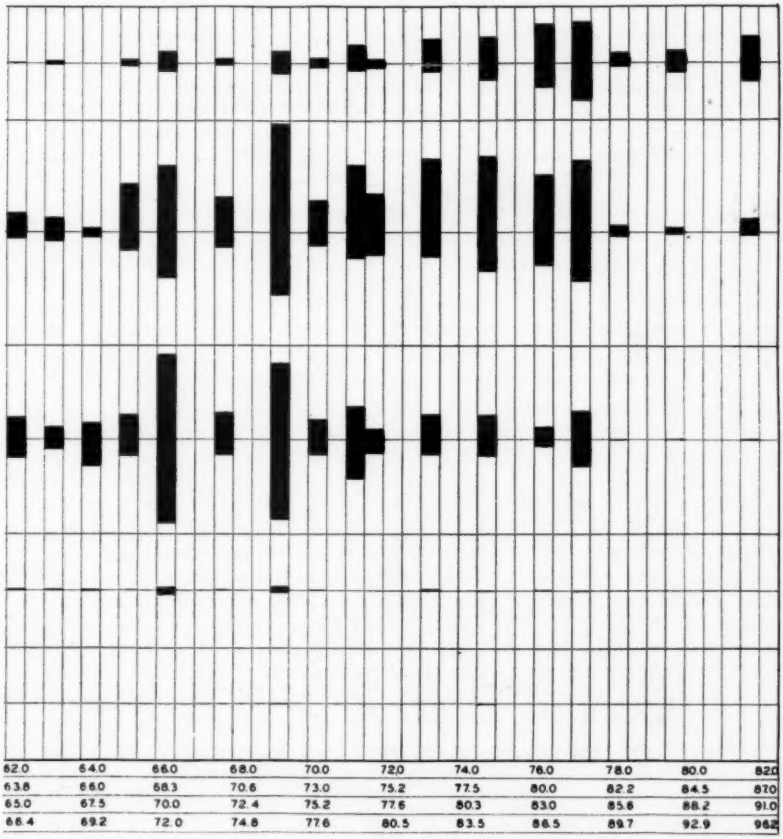

SCALE - NUMBER OF VOTES
SEVERELY WARM WOMEN
SEVERELY WARM MEN
MILDLY WARM WOMEN
MILDLY WARM MEN
IDEALLY COMFORTABLE WOMEN
IDEALLY COMFORTABLE MEN
MILDLY COLD WOMEN
MILDLY COLD MEN
SEVERELY COLD WOMEN
SEVERELY COLD MEN
OUTSIDE DRY BULB AT
EFF. TEMP 44.0 46.0 48.0 50.0 52.0 54.0 56.0 58.0 60.0 E.T.
70% R.H. 43.8 46.0 48.1 50.4 52.5 54.8 57.0 59.2 61.6 F.D.B.
50% R.H. 43.8 46.0 48.2 50.5 53.0 55.3 57.8 60.0 62.5 F.D.B.
30% R.H. 43.5 46.0 48.3 50.8 53.5 55.9 58.5 61.0 63.8 F.D.B.

FIG. 9. RELATION BETWEEN OUTSIDE EFFECTIVE TEMPERATURE AND ACCOMPANYING MEN AND WOMEN EXPERIENCING VARIOUS FEELINGS OF WARMTH

from an outside condition of about 60 deg these same relationships held. After entering from hot outside conditions in the neighborhood of 75 deg the difference between men and women, as regards percentage of ideal comfort, is much less; more men than women felt *mildly cold,* and, while women still predominated in the percentage feeling *mildly warm,* the difference was not great. This indicated that the difference between men and women, as regards their

shock reactions upon entering conditioned space, is less when cooling and air conditioning is applied in hot weather than when heating and air conditioning is applied in the moderately cold heating season.

The curves for both men and women, under and over 40 years of age, show about an eight per cent higher feeling of *ideally comfortable* for those over



DRY-BULB FOR 30, 50 AND 70 PER CENT RELATIVE HUMIDITY AND THE NUMBER OF UPON LEAVING THE INSIDE CONDITION AT 71 TO 72 DEG ET

40 in the outside temperature range from 40 to 45 deg; a lower percentage of feeling of *mildly cold* for men and women over 40, and approximately five per cent higher feeling of *mildly warm* for those under 40. For hot conditions in the neighborhood of 75 deg outside, the figures for *ideal comfort* are reversed: about three per cent more of those under 40 years felt *ideally comfortable*. Likewise the percentage indicating *mildly cold* is reversed with

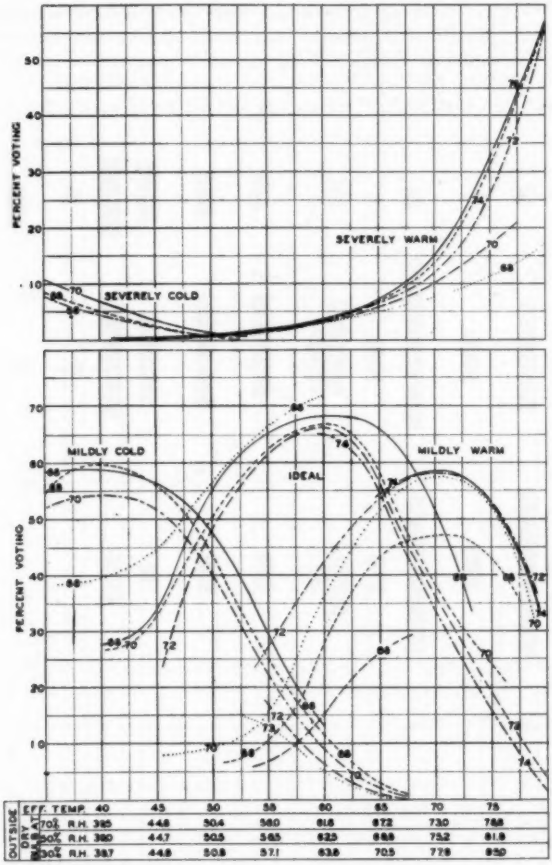

PERCENT VOTING
SEVERELY COLD
SEVERELY WARM
MILDLY COLD
MILDLY WARM
IDEAL
PERCENT VOTING
OUTSIDE DRY BULB
EFF. TEMP. 40 45 50 55 60 65 70 75
70% R.H. 385 448 504 560 618 672 730 788
50% R.H. 380 447 505 565 625 688 752 819
30% R.H. 387 448 508 571 638 705 778 850

FIG. 10. RELATION BETWEEN OUTSIDE CONDITIONS AND FEELING OF WARMTH UPON LEAVING INSIDE CONDITIONS AT VARIOUS EFFECTIVE TEMPERATURES

about five per cent more of those over 40 feeling *mildly cold* than those under 40. The relation between those under and over 40 feeling *mildly warm* is the same as that at lower temperatures: about six per cent more of those under 40 feeling *mildly warm.*

The difference between men and women, in the different age groups plotted in Fig. 8, is not very significant as affecting air conditioning. It is well to point out the fact that the undesirable effects in general are less pronounced with summer cooling and air conditioning than with winter heating and air conditioning. These data bear out other laboratory observations indicating that,

in general, older people are more tolerant of air conditions than younger people. This seems particularly to be the case when objections may be raised due to a feeling of *too warm.*

FEELING OF WARMTH UPON LEAVING AN AIR CONDITIONED SPACE FOR THE OUTSIDE

An example of the tabulated data from all questionnaires answered by individual workers, after leaving the inside space conditioned from 71 to 72 deg ET to the out-of-doors with the prevailing weather, is shown in Fig. 9. Such plots were prepared for each degree range of effective temperature from 66 to 76 based upon a statistical analysis of the data in these 11 charts. The percentage observations for each of the five indices of comfort for several inside effective temperatures are plotted in Fig. 10.

In Fig. 11 the number of reactions weighted for frequency of differential occurrence for each of the five degrees of feeling of comfort is plotted against effective temperature differential between the outside and inside. The reactions are weighted for frequency of differential occurrence by taking the total number of effective temperature differential occurrences for all 16 zones or 16 times the number of days studied, dividing this number by the total number of occurrences of any given temperature differential, and then multiplying by the number of reactions for each of the five degrees of comfort.

Fig. 9 makes possible a visual picture of some data relationships. For outdoor effective temperatures up to approximately 50 deg, there was a maximum of votes for *ideally comfortable,* with votes for *mildly cold, mildly warm* and *severely warm* in descending magnitude. In other words, with the prevailing dress which naturally varies with the outside weather, and with activity incident to leaving the office, a fair percentage enjoyed the outside conditions while lesser numbers felt *mildly cold* or *mildly warm.* For outside temperatures within the range of approximately 55 to 65 deg, the largest percentage was comfortable while numbers in descending magnitude were *mildly warm, mildly cold* and *severely warm.* For outside temperatures within the approximate range of 65 to 80 deg, there is a well-defined decrease in votes for *ideally comfortable,* a much larger percentage increase in votes for *mildly warm* and *severely warm,* and only a negligible per cent indicating *mildly cold.*

Interpretation of the results from the data plotted in Fig. 9 is made difficult by the fact that relatively few observations were obtained for lower outside temperatures and also for extremely high outside temperatures. When these data are further analyzed and plotted as percentages in Fig. 10, comparison is more convenient. The most important facts indicated by the curves in Fig. 10 are the general relationships between the curves for *severely cold, mildly cold, ideally comfortable, mildly warm* and *severely warm.* These curves indicate a marked preference for outside conditions ranging from approximately 50 to 65 deg ET without much regard for the preceding indoor conditions. For lower outside temperatures ranging from 35 to 50 deg, there was a predominate feeling of *mildly cold,* with some *severely cold.* In the approximate range of 65 to 80 deg outside temperature, feelings of *mildly warm* and *severely warm* predominate (*mildly warm* in the lower part of this temperature range, and *severely warm* in the upper part). While these feelings of warmth depend

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more on the outside weather conditions upon entering than on the indoor conditions upon leaving there is a very consistent, though relatively small, variation in the percentage of any feeling of warmth correlating very definitely with the effective temperature maintained in the inside space. As an example, the *ideal comfort* curves show highest percentages after leaving cooler indoor

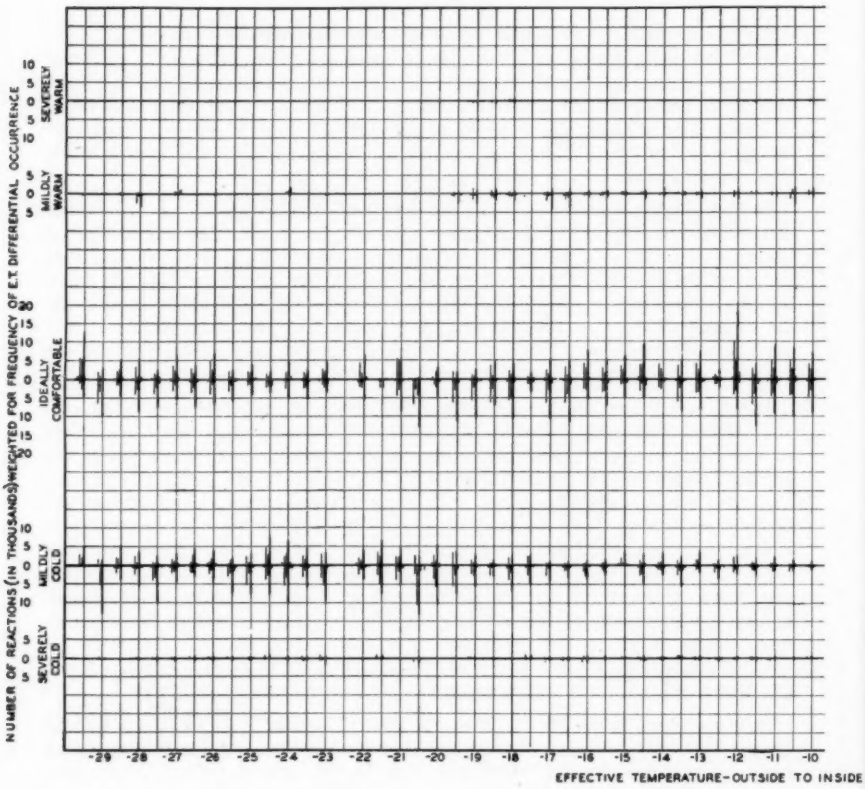
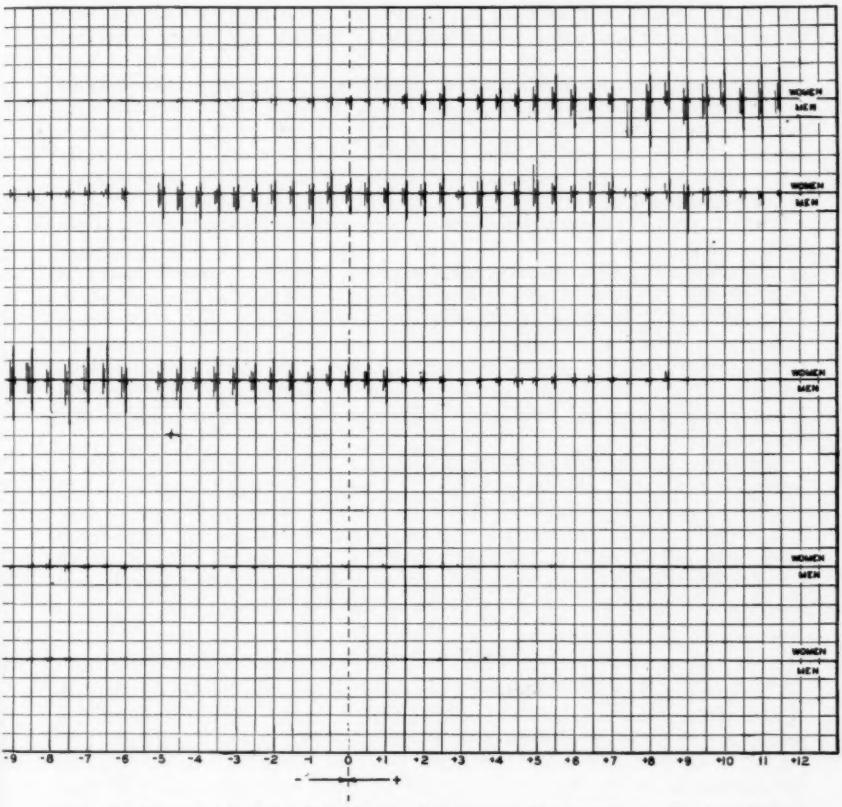

NUMBER OF REACTIONS (IN THOUSANDS) WEIGHTED FOR FREQUENCY OF E.T. DIFFERENTIAL OCCURRENCE
SEVERELY WARM
MILDLY WARM
IDEALLY COMFORTABLE
MILDLY COLD
SEVERELY COLD
-29 -28 -27 -26 -25 -24 -23 -22 -21 -20 -19 -18 -17 -16 -15 -14 -13 -12 -11 -10
EFFECTIVE TEMPERATURE-OUTSIDE TO INSIDE

FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE DIFFERENTIAL BETWEEN OUT- DIFFERENTIAL OCCURRENCE FOR EACH OF THE FIVE DEGREES FEELING OF COMFORT, FOR HEAVY LINES ABOVE OR BELOW THE ZERO LINE GIVES WEIGHTED REACTIONS FOR FOR ALL WORKERS, THE LIGHT LINES FROM LEFT TO RIGHT ON THE LEFT HAND SIDE LINES ON RIGHT HAND SIDE OF HEAVY LINES

conditions within the indoor effective temperature range of from 66 to 74 deg. The same order of percentage holds for *mildly cold* and *severely cold*, but the reverse holds true for the percentage indicating a feeling of *mildly warm* and *severely warm*. Thus for persons entering the outside at 70 deg ET, the percentage feeling *mildly warm* ranges from about 47 per cent for

those entering from an indoor condition of 68 deg to 58 per cent for those leaving at an indoor condition of 74 deg ET. Likewise only 10 per cent felt *severely warm* when they entered an outside condition of 70 deg from an indoor condition of 70 deg, but 15 per cent felt *severely warm* when this same outside condition was entered from an inside condition of 76 deg.



SIDE AND INSIDE AND THE NUMBER OF REACTIONS WEIGHTED FOR FREQUENCY OF PERSONS AFTER LEAVING THE CONDITIONED SPACE FOR THE OUTSIDE. LENGTH OF THE WOMEN AND MEN RESPECTIVELY FOR VARIOUS DIFFERENTIALS; THE HEAVIER LINES OF THE HEAVY LINES UNDER 20, 20 TO 30 AND 30 TO 40 YEARS OF AGE, LIGHT FOR 40 TO 50, 50 TO 60 AND 60 TO 70 AGE GROUPS

The relation between effective temperature differential between outside and inside plotted in Fig. 11 shows a high index of comfort and an approximately equal balance between *mildly warm* and *mildly cool* at a differential of −10 or −11 deg. Since this difference from an average inside condition of 70 or 71 deg ET is obtained at an outside effective temperature of approximately 60

deg, it indicates that the outside temperature rather than the differential is the controlling factor, as shown by Fig. 10.

The relationship between feeling of warmth upon entering an outside condition and the temperature maintained inside indicates that the outdoor condition is far more important than that inside. It also indicates that the condition upon leaving is also significant. However the effect of the indoor condition on the feeling upon entering the outside, as indicated by the curves, does not seem to bear out the persistent claims of laymen and others regarding the reaction after leaving the cooled space. These persistent claims, at least those most often heard, indicate rather dire results in the form of discomfort from heat upon entering the hot outside from an indoor space maintained at too low a temperature. The curves in Fig. 10 show that for hot outside atmospheres a higher per cent of *ideal comfort* is obtained when leaving a cooler inside atmosphere. The percentages of feeling of both *mildly warm* and *severely warm* are lower.

While the bearing upon temperature requirements for air conditioning is not keen, there is some interest in the outside condition giving a high percentage vote of *ideal comfort*. The maximum percentage of *ideal comfort* comes at about 60 deg ET outside, with a high percentage within the range of about 50 to 65 deg, or temperatures considerably below those giving a maximum feeling of comfort while indoors. This is undoubtedly due to a number of factors which include a greater amount of clothing worn while out-of-doors, the effect of sun radiation, the greater amount of physical exercise, the mental stimulus upon leaving the office, and the general pleasure accompanying release from restraint.

Most important in this phase of the study is the almost complete lack of increased unpleasant reaction to leaving a cooled space with decrease in the maintained indoor temperature. That this is real is indicated by a more general analysis of the reactions of persons leaving cooled and air conditioned space contained in another Laboratory study, to be published later, in which a large percentage of more than 250 persons indicated no general unpleasant reactions following their exit from cooled and air conditioned space.

ACKNOWLEDGMENT

The Research Technical Advisory Committee on Sensations of Comfort is indebted to the Minneapolis-Honeywell Regulator Co. for their cooperation in providing the facilities of their general office building and the necessary personnel to observe and collect the data for these investigations, and to the Brown Instrument Co. for furnishing the recording instruments used in these studies.

(See Joint Discussion, p. 566)

No. 1103

GENERAL REACTIONS OF 274 OFFICE WORKERS TO SUMMER COOLING AND AIR CONDITIONING

By F. C. HOUGHTEN,* A. B. NEWTON ** (*MEMBERS*), R. W. QUALLEY,*** AND EDWARD WITKOWSKI † (*NON-MEMBERS*)

This paper reports the results of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Committee on Research, conducted under the supervision of the Technical Advisory Committee on Sensations of Comfort and analyzed by the Research Laboratory.

THIS paper deals with the results of a questionnaire sent to 274 employees working in air conditioned offices in Minneapolis during the summer of 1937. These employees were the same as those participating in the daily study reported in the two earlier Laboratory reports.[1, 2]

The questionnaire was given to the employees on September 15 after they had been accustomed to filling out the regular daily questionnaires included in the studies referred to since May 4. Therefore, those participating had more than the usual background in studying and interpreting their reactions to air conditioning, but with the exception of a few of the higher ranking members of the organization who may have been included in the survey, they had no particular knowledge or interest in air conditioning. It is believed that the results of this questionnaire will serve a useful purpose in indicating the reactions of normal persons occupying air conditioned space during the working part of the day, and that they will help clear up a number of questions which are generally conceded to be controversial in the minds of many laymen and some persons more closely associated with air conditioning.

A copy of the questionnaire which was given to employees on September 15 and collected on September 16 is included. No other advice or suggestions were given concerning the returns to be made.

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† Student Assistant, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.

[1] Summer Cooling Requirements of 275 Workers in An Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337.)

[2] Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley, and M. C. W. Tomlinson. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 569.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938, by A. B. Newton.

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QUESTIONNAIRE ON AIR CONDITIONING

(Answered by 274 Office Employees)

Your cooperation in filling in the daily questionnaire cards, telling us of your reactions to the comfort conditions obtained by the office air conditioning system is very much appreciated. We have obtained a great deal of valuable data from these daily cards, and as we near the end of the cooling season, we also desire your answers to the following questions. We should appreciate as much detail as you can supply:

(1) What difference, if any, have you noticed in your reactions after leaving the office on warm days as compared to the period previous to the installation of air conditioning?

(2) What difference have you noticed between your reactions after leaving the office and reactions of friends or associates who were not in conditioned offices during the day?

(3) General remarks on the effectiveness of air conditioning.

A preliminary analysis of the returns showed certain marked tendency in the replies and included several definite types of answers to each of the three questions. All of the replies are tabulated under the headings of the three questions according to the type of answers.

IMMEDIATE REACTIONS UPON LEAVING THE AIR CONDITIONED SPACE

Replies to Question 1, What difference, if any, have you noticed in your reactions after leaving the office on warm days as compared to the period previous to the installation of air conditioning?, were divided into four divisions and sub-divisions:

(1) Felt too warm for short period after leaving.
  (a) Less than 20 min.
  (b) Less than one hour.
  (c) Few hours.
(2) Felt too warm all night.
(3) Felt indifferent to warmth after leaving.
(4) No intelligent answer.

While different persons showed wide variation in the length and character of their statements they could with few exceptions be fitted into these divisions. The percentages given each type of answer to Question 1 are plotted in the chart, Fig. 1. Two hundred seventy-four questionnaires were sent out but only 243 answered this question. The others were not present before the installation of air conditioning; therefore the percentages are based on the 243 answers. They show that 35 per cent generally felt too warm for a period of from less than 20 min to two hours after leaving the cooled space, and that approximately 13½ per cent felt too warm throughout the entire night. Another 35 per cent indicated that, in general, they had no noticeable reactions to the hot outside or that they felt indifferent to the outside weather after leaving the conditioned space. Approximately 17 per cent of the replies were not sufficiently to the point to be tabulated in any of the groups.

Many of the answers to Question No. 1 showed very definite characteristics in the feelings of the employees after leaving the air conditioned space. Some

of these are of interest in forming a better understanding of air conditioning. A smaller percentage are of interest because they are exceptions. In other cases, the replies were amusing but still educational. A number of typical replies without respect to the percentages of each are quoted:

Was acutely aware of the high temperature immediately upon leaving the building.

A feeling of being quite warm for approximately one-half hour with normal reactions thereafter.

I noticed very little difference. It seemed slightly warmer at first but after being out an hour or so everything was okey.

The sensation of fatigue upon leaving the office has been less noticeable since the installation of air conditioning.

That tired, worn-out feeling on hot summer days is eliminated and I think it enables one to accomplish much more work and still feel comfortable.

I seem to mind the outside heat more. My joints seem to pain me, especially my leg joints,—something like a rheumatism.

Previously the air outside seemed clean and fresh compared to that of the factory.

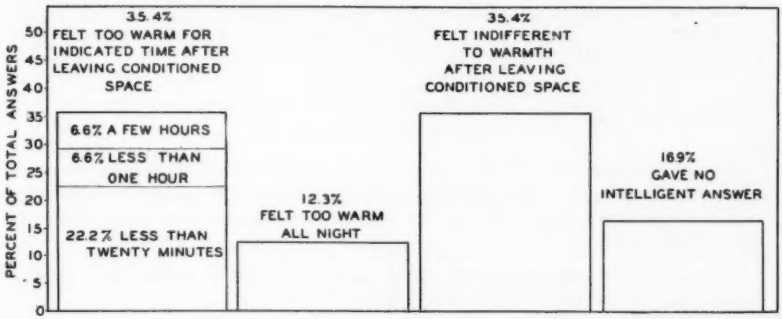

FIG. 1. ANSWER TO QUESTION NUMBER ONE. WHAT DIFFERENCE, IF ANY, HAVE YOU NOTICED IN YOUR REACTIONS AFTER LEAVING THE OFFICE ON WARM DAYS AS COMPARED TO THE PERIOD PREVIOUS TO INSTALLATION OF AIR CONDITIONING?

Now, after working in an air conditioned space, the outside air seems stuffy, hot and full of dust and dirt.

On excessively hot days I perspired and noticed the heat more than the rest of the family in the early parts of the evening. In fact, I believe it took about four hours for me to become acclimated to the temperature rise.

I noticed quite a change in temperature upon leaving the office on warm days; this change lasted for a short period. During the time previous to the installation of air conditioning my reactions were very much the same upon leaving the office as they were to the air conditions in the office.

The first few months of operation of the system made one feel comfortable inside during the day but in the evening the extreme change of temperature (75 to 100 deg) from inside to outside nearly suffocated one when experienced all at once. But the last few months of raising the inside temperature in the late afternoon (3:00 to 4:30) has made a considerable improvement.

On warm days there is a reaction of being severely warm when one goes from indoors to outdoors, but this is offset by the ability to withstand the heat outdoors better because one is not nearly as fatigued as one would be where no air conditioning is installed. A person can stand the heat better if he is not all tired out from working in the heat all day.

I have not been nearly as fagged out during the hot evenings as I was before the installation of air conditioning.

After leaving an air conditioned office, one is better prepared to withstand higher

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temperatures outside than if one has been subjected to high temperatures throughout the day.

Before the air conditioning system was installed, working at a drawing board on warm days was very difficult, because perspiration on the arms and hands would spoil a drawing. Now, one does not have this condition to contend with and more work can be accomplished with no feeling of being tired.

I did not feel all worn out by the heat during the day, and although the heat was very noticeable a short time after leaving, I became accustomed to it and was more able to take it as I was not all tired out.

No difference, except that undoubtedly on warm days the outside weather momentarily seems warmer when leaving conditioned space than it would in going outside from unconditioned space.

I seemed to notice the heat more on excessively warm days after leaving a conditioned space than I did while working in an unconditioned space. This sensation of heat lasted most of the evening and night on very warm days.

Having worked in an air-conditioned office all day, the breeze in the evening would appear to be warm or hot. After working in unconditioned offices, and perspiring freely, any evening breeze would appear to be cool.

My skin is so cool that the heat doesn't affect me very greatly.

Heat was very intense and usually lasted until it cooled off after sundown.

The evening heat is less noticeable, because my system has not been under the pressure of heat during the warmer hours of the day.

REACTIONS AS COMPARED WITH THOSE WHO DID NOT SPEND THE DAY IN AIR CONDITIONED SPACE

Replies to Question 2, What difference have you noticed between your reactions after leaving the office and the reactions of friends or associates who were not in conditioned offices during the day?, were divided into three divisions and sub-divisions:

(1) Stood hot weather better than if had not spent days in cooled space, or than friends who had not spent days in cooled space.
  (a) Less complaint of heat.
  (b) Less fatigued, less tired, or had more pep.
(2) Did not stand hot weather as well as if had not spent days in cooled space. Friends, who had not spent days in cooled space, stood heat better.
(3) No difference in way they withstood hot nights compared to like feeling when days were not spent in cooled space, or when compared with feelings of friends who did not spend days in cooled space.

Again wide variations in the nature and characteristics of the statements were found in the replies, but with few exceptions the replies could be included in the groupings as given. The percentage showing each type of answer is given in the chart, Fig. 2. Two hundred seventy-four questionnaires were sent out, but only 226 answers to the question were received. Therefore percentages are based on the 226 answers. A little over 70 per cent indicated in one way or another that they were not as fatigued during hot spells as they would have been had they not worked in air conditioned space during the day time, or as compared with their friends who did not work in air conditioned space. The word *fatigued* naturally only occurred in a relatively small percentage of the total number of replies, but other expressions such as *had more pep, felt less washed-out,* or *less let-down* were included in this grouping. Approximately 13 per cent indicated in one way or another that they did not stand the hot weather as well as they would if they had not spent their time in air conditioned space, or as their friends and associates who did not work in air

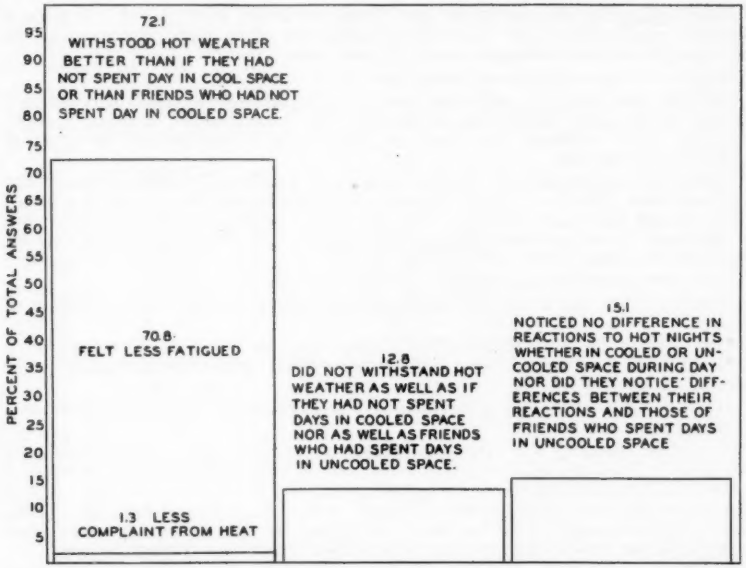
72.1
WITHSTOOD HOT WEATHER BETTER THAN IF THEY HAD NOT SPENT DAY IN COOL SPACE OR THAN FRIENDS WHO HAD NOT SPENT DAY IN COOLED SPACE.
PERCENT OF TOTAL ANSWERS
95
90
85
80
75
70
65
60
55
50
45
40
35
30
25
20
15
10
5
70.8
FELT LESS FATIGUED
1.3 LESS COMPLAINT FROM HEAT
12.8
DID NOT WITHSTAND HOT WEATHER AS WELL AS IF THEY HAD NOT SPENT DAYS IN COOLED SPACE NOR AS WELL AS FRIENDS WHO HAD SPENT DAYS IN UNCOOLED SPACE.
15.1
NOTICED NO DIFFERENCE IN REACTIONS TO HOT NIGHTS WHETHER IN COOLED OR UNCOOLED SPACE DURING DAY NOR DID THEY NOTICE DIFFERENCES BETWEEN THEIR REACTIONS AND THOSE OF FRIENDS WHO SPENT DAYS IN UNCOOLED SPACE

FIG. 2. ANSWER TO QUESTION NUMBER TWO. WHAT DIFFERENCE HAVE YOU NOTICED BETWEEN YOUR REACTIONS AFTER LEAVING THE OFFICE AND THE REACTIONS OF FRIENDS OR ASSOCIATES WHO WERE NOT IN CONDITIONED OFFICES DURING THE DAY?

conditioned space. These answers again were cast in various forms, including statements that they *were more fatigued, more tired, had less pep,* etc. Approximately 15 per cent indicated no marked preference to air conditioning as regards their feelings during the evenings and nights of warm weather. A few characteristic answers which may be of interest are given:

In general, I was more comfortable, and usually was not quite as tired as a person working in a non-air-conditioned office.

Most people complained about being all tired out and weakened by the heat, but I never had such reactions.

I noticed their apparent consciousness of the heat during the day, whereas, I had not been aware that the day had been uncomfortably warm.

The first hour after leaving the air-conditioned office on a hot day I noticed the heat but soon became accustomed to the warmer air. I noticed friends and associates showed the effects of the hot weather more when they were in an un-conditioned building all day.

They don't mind the outside heat as much as I do. They have as a general rule more pep than I have.

I couldn't talk about the heat of the day. They seem to be more worn out.

I believe I can truthfully say that most of my friends who were not in air conditioned places during the heat of the day were practically worn out by night, while I, being in an air conditioned space all day, was still feeling fine regardless of the perspiring mentioned heretofore.

In general there has not been the complaint this summer as voiced in previous summers. However, the greatest complaint has been poor wives who are domestically confined during the period in which their husbands are air conditioned.

After leaving the office on warm days I was not nearly as tired and I had more ambition; acquaintances who were not in conditioned offices were more tired out at the end of the day and seemed to mind the heat more.

People in non-air-conditioned places were more tired.

I felt the heat more, but since they had to work in it all day, it actually affected me less, in the matter of physical well-being, than it did them. I acquired a slight heat-rash on hot evenings, for about two weeks, but it was probably a condition aggravated by the heat.

I think that my friends who did not work in air conditioned buildings minded the heat more than I did after leaving the office. Air conditioning leaves one in a better frame of mind after leaving work because of being comfortable all day. Persons who do not work in air conditioned places think about the heat all day and by evening they think it's much hotter than it really is.

They were very tired and irritable whereas I felt fine.

I find that I am not as easily irritated as those who do not have the advantage of air conditioning.

After leaving the air conditioned office and returning home, those outside were tired out from the constant heat, while those working in the air conditioned room were still fresh.

At quitting time they were all tired out, not from the work but because of the heat all day. I was not.

In general, reactions were the some except on very hot days. Then friends did not seem to mind the heat as much as I. About 3 or 4 hours in the evening I noticed a difference in reactions.

A definite difference—on hot days followed by hot, close, muggy evenings—I have been much more comfortable and have complained far less of the heat than other members of the family who have not been in conditioned space during the day.

Friends appeared to be every bit as hot and uncomfortable as I was,—in fact, they suffered all day while I was glad to be in the conditioned office.

After a day of comfort in the office, hearing people groan on the street cars going home about the terrific discomfort they had endured at their places of employment, the value of the privilege of spending the day in air conditioned quarters was emphatically indicated. At home, likewise, the complaint of intolerable conditions that made it impossible to accomplish anything during the day was proof of misery we had been spared.

The heat affected them more than it did me. They had made themselves warmer by fretting about the severe heat.

I perhaps noticed the heat outside more than the person who had not been in a conditioned office during the day, but was less tired and could more easily endure the heat for the rest of the day.

Apparently answered previously. I might add that I am more able to stand the heat of a warm evening than those who have been extremely warm all day.

GENERAL REACTION TO AIR CONDITIONING

Replies under the heading of Item 3, general remarks on the effectiveness of air conditioning, indicated a logical division into three divisions and subdivisions:

Analysis of replies to Question No. 3:

(1) General reaction to air conditioning good.
  (a) More comfort and pleasantness.
  (b) Better efficiency.
  (c) Free from colds.
  (d) Better feeling or more pep.
(2) General reactions to air conditioning not good.
  (a) At times, too cool or drafty.
  (b) At times, too warm.

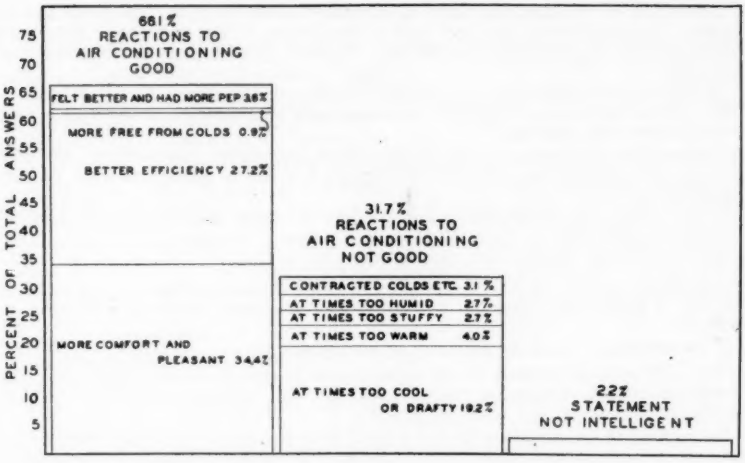

FIG. 3. GENERAL REMARKS ON EFFECTIVENESS OF AIR CONDITIONING

(c) At times, stuffy.
(d) At times, too humid.
(e) Contracted colds or other derangement, including rheumatism or body pains.

(3) Statement not intelligent.

The usual wide variations in the answers were found, but they could be divided into the divisions and sub-divisions given. The chart, Fig. 3, gives the percentages of replies under each classification. Two hundred seventy-four questionnaires were sent out, and 224 statements of reactions to air conditioning were received. Percentages are based on the 224 replies. Approximately 66 per cent showed a satisfactory appreciation for air conditioning. These were sub-divided into approximately one per cent indicating a greater freedom from colds, approximately four per cent having a general feeling of greater satisfaction or more pep, approximately 27 per cent indicating a greater efficiency in their work according to their own interpretation, and approximately 34 per cent spent a generally more comfortable and pleasant summer. Approximately 32 per cent showed a lack of satisfactory appreciation for air conditioning or had a definite criticism of the same. Of these approximately three per cent indicated that conditions were at times too humid, approximately the same percentage that the conditions were at times too stuffy, another three per cent indicated that they were more subject to colds than previously, and four per cent complained that the air conditioned space was generally kept too warm. Of the 32 per cent unsatisfied with air conditioning, there was a rather general complaint covering approximately 19 per cent that the air conditioned space was too cold or drafty. Approximately two per cent answering Question 3 gave answers which were unintelligible.

A few characteristic replies to this question are given:

Appreciate air conditioning only in extremely warm weather; otherwise far too drafty. Further, it does not take care of the smoke in the engineering department which irritates my eyes, nose, and throat. I believe open windows would prove more beneficial.

Was bothered by hay fever out of office. This condition was relieved when in the office. I feel that an air conditioned office has made it possible for me to have a much more comfortable summer.

I think it is very good during warm weather and severe cold weather, but when it is so ideal outside that you want to stay out I prefer to be able to open the windows and enjoy it.

It is a great help to non-smokers who must work between inveterate smokers all day.

I have noticed that the office has been rather drafty at times.

Had to keep my coat on to prevent myself from sneezing.

Air conditioning is okey, except along the south windows,—when the sun shines in here it is hotter than outside. I suggest that you turn on more cold along the windows.

Excellent—keeps personal efficiency at maximum.

My own personal opinion of the effectiveness is that it at least keeps you comfortable one-third of each day and it most certainly is great relief from hot spells in the weather which last weeks at a time.

The effect of air conditioning is beneficial in many ways. It adds greatly to the amount of work that can be produced, especially in the hot summer. It keeps everybody happier and more agreeable. It helps one's health as one is better able to fight off sickness when not all tired out. It offers relief from hay fever and other nose and throat irritations.

The general office temperature is held too low for the comfort of one employed at a desk all day.

After leaving the air conditioned office and entering the warm unconditioned places, perspiration began to flow freely, and for a short time a sort of dizzy feeling would be felt, but would soon wear off.

A person could work all day at the desk without feeling tired or sleepy. No danger of spoiling tracings and papers by excessive perspiration. I think a person working in air conditioned space during the day should try to get into the sun and outdoor heat as often as possible in the evenings, Saturdays and Sundays to induce natural perspiration, as I noticed I felt better the summer of 1937 than 1936 when I tried to avoid the heat after working hours.

In spite of the excessive reaction of heat in the evening, air conditioning is far better than no conditioning, but it could be improved by elimination of draftiness and a certain odor which seems to result from the outdoor air.

I will not soon forget the discomfort of unconditioned space affording only a choice of two evils—perspiration or exasperation—the latter with a fan blowing one's hair into utter disorder and scattering papers. What ideal air conditions mean to health and comfort and efficiency and enjoyment cannot be condensed into so little space. The passing of the day of drafts from open windows or suffocation with them closed, afternoon fatigue from working in atmospheres lacking proper humidity, air circulation, or at a temperature too high or too low, should mean much to progress and accomplishment for everyone. Summing up, the benefits in health and comfort are immeasurable. With what pride couldn't we show our accomplishment of the optimum air condition to that fellow, Mark Twain, who said, "Everybody talks about the weather, but nobody does anything about it."

Easier to work when office is air conditioned during summer. Would be better if it could be the same in temperature everyday.

More work accomplished during the day than previously and with less fatigue.

More effective work can be done when there is no (or but little) distracting air condition. Due to large effect of many variables—personal health conditions, drinking of water, type of clothing, etc.,—the shock is a delicate, indefinite and negligible factor; too elusive to be pinned down quantitatively. I am, simply, comfortable in conditioned space and uncomfortable in extreme conditions.

DISCUSSION OF RESULTS

In general, the answers to the three questions presented in the questionnaire show a considerable percentage of general satisfaction in the experiences resulting from air conditioning. Better than 35 per cent were frank in stating that they felt too warm for a while after leaving the air conditioned space. However, the duration of the hot reaction was in most cases not great, and no great importance was placed on it. An equal percentage showed an entire indifference to or a lack of this hot reaction. Only approximately 13 per cent of the total indicated that their evening and night reactions to the heat may have been increased due to their experience with air conditioning during the day. Undoubtedly, a considerable portion of these lacked satisfactory background on which to base their interpretations. In other words, at least 75 per cent of all inquiries, and a larger percentage of those giving intelligible answers, had no overwhelming or serious reaction upon leaving the conditioned space.

Again, a little over 70 per cent indicated clearly that their unpleasant reactions to the hot nights or unpleasant reactions to a spell of hot weather were lessened by spending the working part of the day in the summer cooled and air conditioned space. When 15 per cent of those who were indifferent to the heat of the night are added to these, less than 13 per cent are found who had a definitely unsatisfactory reaction to the hot evenings and nights, which was charged to air conditioning.

Sixty-six per cent in one way or another indicated definite positive reaction. These reported better feeling, more pep, greater efficiency, greater comfort and pleasure due to spending the working hours or hot portion of the day in air conditioned offices. While it is true that almost 32 per cent stated some objection, these objections were usually limited to certain times, and should not, therefore, be definitely interpreted as a general objection to the satisfaction of air conditioning. They should be accepted rather as objections to occasional lack of perfection which may have actually existed at times in some parts of the extended office space. In this connection, it should be emphasized that on certain days during the study the conditions in small portions of the offices were actually upset so as to make them too warm or too cool for maximum comfort, in order to obtain data for such conditions. It is quite possible that some of these definite conditions which only pertained for short periods of time made more impression on certain individuals than the general satisfaction which they may have felt toward air conditioning in general.

It is felt that the questionnaire serves a useful purpose in indicating the large percentage of general satisfaction to air conditioning and the lack of severity and importance of the hot reaction upon leaving the air conditioned space. It is logical that this should be the case, since it is generally recognized that the unsatisfactory reaction of the human body to an extended hot spell is an accumulative effect. It is well recognized that a person can spend a few hours during the day in a very warm atmosphere without any extended effect. In other words, recovery from short exposure to high temperature is rapid and more or less complete. On the other hand, exposure to hot atmospheres during a hot spell in the summer when one is unable to avoid the heat either night or day is severe and accumulative. The answers to these questionnaires go a long way to satisfy this contention. The questionnaire offers a little data on the contention that a high percentage of cold infections results from

exposure to low temperatures maintained in air conditioning. Approximately one per cent indicated a greater freedom from colds, while approximately three per cent indicated a greater incidence to colds. Since it is generally well recognized that summer colds are usually due to infections rather than exposure, it is probable that the few replies received in this connection are of little importance.

The one reply, listed in the sample replies to Question No. 3, charging the feelings of pains in joints and muscles, or rheumatism, to air conditioning is probably another case of the exception which proves the rule. It is probable that in any such extensive group of workers one or two could be found who had recently contracted some form of rheumatism, lumbago, or other source of muscular or joint pains.

Similar studies to those carried on in Minneapolis during 1937 are being planned for other sections of the country during the summer of 1938. The importance of this type of analysis of occupancy in air conditioned space is just being recognized and should be made an important part of future investigations.

ACKNOWLEDGMENT

The Research Technical Advisory Committee on Sensations of Comfort is indebted to the Minneapolis-Honeywell Regulator Co. for their cooperation in providing the facilities of their general office building and the necessary personnel to observe and collect the data for these investigations, and to the Brown Instrument Co. for furnishing the recording instruments used in these studies.

(See Joint Discussion, p. 566)

In Memoriam

Name	Joined the Society	Died
George H. Blanding	1918	May 1938
William P. Boddington	1927	Nov. 1938
Francis J. Currie	1935	Apr. 1938
Thomas K. Dobie	1935	Mar. 1938
Henry S. Downe (*Charter and Life Member*)	1895	Dec. 1938
Frank C. Gibbs	1920	July 1938
John A. Gross	1937	June 1938
James D. Hoffman (*Presidential and Life Member*)	1903	Aug. 1938
Edmund T. Keefe	1931	Dec. 1938
William E. Kemery, Jr.	1937	Jan. 1938
Peter F. Maginn	1908	Feb. 1938
Liguon A. McGuigan	1919	Sept. 1938
William H. McKiever (*Charter and Life Member*)	1895	Aug. 1938
John McKinley	1938	Dec. 1938
Edward A. Munro (*Charter and Life Member*)	1895	May 1938
Walter A. Oates	1931	July 1938
John H. Plunkett	1925	Jan. 1938
A. Kirk Purdy	1922	Jan. 1938
Herman Seelbach	1931	Dec. 1938
Warren Webster, Sr. (*Life Member*)	1899	Dec. 1938
Howard A. Wood	1937	Sept. 1938

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